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FUTURE HORIZONS IN DIESEL ENGINE DESIGN.

SPEECH DELIVERED BY

CAPTAIN L. F. SMALL, U. S. NAVY

BEFORE SAE MEETING (DIESEL) IN

CHICAGO, MAY 17, 1944

This war will be known to history for its amphibious operations and American engineers and industry will long remember it for the millions of internal combustion engines which have been built already and will continue to be built in ever increasing numbers. They provide that flexibility and radius at sea, through the air and over the ground, which will eventually overwhelm our enemies.

In congressional hearings of last December, the Navy announced the completion of some twenty-five thousand landing craft and obtained funds for the 1944 amphibious program which comprises even larger numbers involving in all about twenty-

seven million, five hundred thousand horsepower in Diesel engines for their propulsion.

It has been the task of the Bureau of Ships to assume responsibility for the proper utilization of all these engines, for their construction by practically every responsible engine builder in the country, and for their operation and maintenance in every corner of the globe by every type of operator who has been hastily trained for the job. From this vantage point, I wish to compliment most heartily and without reservation the splendid cooperation of American engineers and manufacturers in producing the heretofore undreamed of totals of horsepower. The teamwork of the worker, the engineer and management has been magnificent and will remain forever an achievement of the highest caliber.

However, the end of the war is not yet in sight. If we are to reduce the bloodshed and the loss of life, efforts of still greater magnitude on the part of all of us are absolutely essential. It is inconceivable to me that we have done our best already, since that would be equivalent to saying that the future holds no promise to men of courage and imagination. I am proceeding therefore, on the assumption that you men need only a clear, statement of what the Navy could use to advantage in this war and in the future. I assume that you are the true disciples of Archimedes and Erickson, to name but two of the engineers whose work for the war effort of their countries changed the history of mankind.

The United States Navy clearly defined its position with respect to internal combustion engines when it authorized Diesel power boats for all new cruisers and destroyers of the 1933 construction program. Subsequently, the production of gasoline engines for power boats was stopped completely at the Norfolk Navy Yard and since then, over five thousand Diesel engines have been built there for standard ships boats alone, with several thousand others procured from a private builder. The war has justified fully the Navy's choice of the Diesel engine for its unsurpassed fuel economy and absence of fire hazard. Many types of Diesel engines have been developed since to propel and power a variety of naval vessels and at least one manufacturer completed over twenty thousand engines of the same size and type during 1943.

The Navy believes wholeheartedly in Diesel engines and I am

proud to announce that as of October, 1943 the combined horsepower of Diesel engines being installed in Naval craft exceeded that of the steam turbines for the first time in the history of your Navy. Considering the fact that every time a capital ship or aircraft carrier is commissioned over two hundred thousand, and with each destroyer over sixty thousand horsepower in steam turbine output is added, while Diesel engines power ships of less than twelve thousand horsepower, this interchange in rank and importance between Diesel and steam is a highly significant milestone. I am glad, therefore, to make this announcement amid the multitude of happy contributors to this accomplishment.

What is it then that keeps the Diesel engine from use in combatant surface ships of the Navy and how could such a goal be achieved? The answer to this question, based on the Navy's extensive experience, may be helpful to all engineers interested in future development of the engine.

The Navy appreciates the instant readiness of the Diesel engine to take full load for it means quick getaway for a ship at anchor and ability to increase speed at will while underway should occasion demand it. Such disasters as Oran, Taranto and our own Pearl Harbor would have been greatly lessened had the ships been propelled by Diesel engines. On the other hand, the size of the propelling plants, ranging from sixty thousand to over two hundred thousand horsepower, is manifestly out of the range of existing designs of Diesel engines. What is needed, therefore, are units of much higher output, of lighter weight, smaller bulk and much simpler construction.

Every engineer understands that every individual turbine blade is a small, but, nevertheless, complete steam engine unto itself for converting the heat energy of steam into the mechanical energy of the rotating turbine shaft. Yet, it is simplicity itself—a cantilever beam of mathematically determinable cross-sections with an elaborately shaped root for fastening to the turbine disk, but a single piece nevertheless. Is it a wonder then that as many as two hundred and fifty thousand blades per ship had been installed in some Naval vessels without hesitation?

Much has been done already to widen the scope of usefulness of the Diesel engine in the Navy. Representatives of the Bureau of Ships on several recent occasions have traced its development

since the last war, when twenty-nine cubic inches of piston displacement were required for each horsepower output, to our latest types requiring only two and one-half cubic inches. The reduction of weight has been from sixty pounds per horsepower to four and one-half pounds with consequent shrinkage in bulk. The development of welded steel construction and production of many ingenious arrangements of cylinders have also been reported on. Such current problems as the need for simplified treatment of torsional vibration and for development of uniform and objectively conceived standards of acceptable porosity in the chrome-plating of cylinder liners have been brought to this society, and I am happy to report your wholehearted cooperation through the medium of the War Engineering Board. Two committees of outstanding authorities in the field have been organized and practical results doubtless will be forthcoming very soon.

The performance of the Diesel engines in the Navy is most gratifying. No engineer will construe this to mean that we have no troubles, for that is impossible as long as human beings build the engines. However, for every engine which gave us trouble, we have scores and even hundreds of its dulpicates, in some cases, that operated trouble free for thousands of hours. It is true that some engine types are more foolproof in operation than others and some designs will stand more abuse and neglect, such as are inevitable in the grim business of war and result from the natural impulse to fight for survival. But I repeat that on the whole, the Diesel engines in the United States Navy have been performing satisfactorily and have earned the confidence of all operating personnel. Hence, anything that I will say from now on must not be construed as a criticism of the equipment with which the Diesel industry has provided us, but as a sincere attempt to define the road of highest promise into the future.

To begin with, I see no room in the future for the four-cycle naturally aspirated engine. During two strokes of the four-stroke cycle, the cylinder pressures are substantially atmospheric—hence, one-half of the weight of this engine must be considered as the weight of its charging equipment. In a two-cycle engine, the weight of its scavenging blower is manifestly very much lighter than the weight of the engine proper and accounts for the weight and bulk advantage of the two-cycle engine. We can all agree that the four-cycle naturally aspirated

engine is a marvel of simplicity of construction, its pistons and valves are free from heat troubles and it will, no doubt, survive for long in our automobiles, but we cannot afford such a luxury in a Naval vessel where weight and space are at a premium and badly needed for the primary purpose of a fighting ship. The use of a high-speed blower of light and simple construction to supercharge the four-cycle engine appears inevitable to me for future designs.

Economic necessity will force us to take advantage of every metallurgical development for parts exposed to heat in order to increase the M. E. P. and horsepower output of the engine and thereby reduce its weight and bulk. This will result in higher pressure within the cylinder at the moment exhaust begins and consequently in greater losses to the exhaust gases with higher fuel consumption and lower efficiency unless we utilize the energy in the exhaust gases. Buchi pioneered in this field and achieved roughly a fifty per cent increase in the horsepower output of a four-cycle engine by expanding its exhaust gases in a turbine driving a centrifugal blower, which thereby supplied over fifty per cent more air to the engine for combustion of increased amounts of fuel. Furthermore, Buchi turbocharged engines have lower exhaust temperatures, and, therefore, greater margin of reliability for pistons and valves. The fuel consumption of these engines is greatly improved also. Buchi's success rests upon his patented grouping of cylinders exhausting into one manifold to secure non-interference. However, his own published tests show that greater energy will become available to the turbine if every cylinder has its own exhaust duct to the turbine. This arrangement provides higher supercharging pressure with consequent further increase in engine output. Improved turbocharging by the use of individual exhaust passages appears to be the next step which builders of four-cycle engines should adopt if this type of engine is to survive the competition of the post-war period.

Considerable numbers of several different types of high-speed two-cycle engines are in service in the United States Navy. While all of them perform with varying degree of success, I have no hesitation to make the statement that the future belongs to unidirectional scavenging types and I will even go one step farther and say very much more effort should be concentrated on the development of the opposed-piston type of engine, where

each of the two pistons is used to control a row of ports—scavenging and exhaust. The compelling reason for making this bold statement concerns the simplicity of construction of this type of engine and the greatly reduced number of parts, with the accompanying reduction in noise due to non-existence of poppet valves. In my opinion, the poppet valve must be abandoned if we are to have engines suitable for propulsion of combatant surface ships. The poppet valve is the principal barrier to higher engine speeds because it is the main source of noise and vibration and the bottleneck in the path of escaping gases. Valves, seats, springs, rockers, camshaft and its drive—all of these will vanish together to the profound satisfaction of the operating personnel.

All of this is really not a very original deduction on my part. As time goes on, many more engineers learn to appreciate the advantage offered by the opposed-piston design and several different variations have been developed in addition to the original design with a single crankshaft and side-rods. Sulzer Brothers has recently adopted the two-crankshaft opposed-piston construction, developed by Junkers for aircraft engines. In addition, Sulzer Brothers have been building a horizontal rocking-lever type of opposed-piston engine for the last few years. Burmeister & Wain, as well as Harland & Wolf, have finally adapted the side-rod opposed-piston design to their double-acting two-cycle engine after trying several designs with poppet valves for exhaust. Finally, several different designs of crankless engines, now in the course of development, also use the basic opposed-piston feature and there is no reason to assume that additional and better adaptations are impossible.

With high capacity ports for both intake and exhaust, each cylinder becomes a simple pulsating combustion chamber scavenged and recharged after each charge of fuel is burned, with compression used to produce auto-ignition temperatures and expansion of gases to yield power. The expanded exhaust gases mixed with the surplus scavenging air will constitute the new low-pressure energy medium to be expanded finally in a turbine of much higher rotative speed and, therefore, of lighter construction with consequent lowering of weight per horsepower for the complete unit.

It may appear to you that we have arrived at something all practical men have frowned upon ever since Diesel himself failed

in the attempt to make it practical; namely, compounding, which has fascinated also such distinguished engineers as Junkers, Sperry and George Curtis of the turbine fame. But be it as it may and whatever the label of designation, the time, in my opinion, has arrived when practical men responsible for powering of ships must prod the industry into the next step of development because the successful performance of the elements necessary for the next type of internal combustion prime mover warrants it. We must move forward.

Now—shall we head immediately to the gas turbine as the sole generator of torque or is there some intermediate approach? I do not believe it is important that this question be answered in either affirmative or negative at this time. Suffice it to say that the turbine must be invited to play as big a role as it is capable of doing successfully. Its ability to utilize exhaust gases of four-cycle engines effectively has been demonstrated. The decision as to the line of demarkation between the engine and the turbine as well as to the level of pressure and temperature of the gases for the turbine must be made in the near future. It is safe to state that initially we can assign the turbine the same task it is so admirably performing for the four-cycle engine; namely, that of providing scavenging and charging air for the two-cycle engine. Why was it not done long ago, you may ask. The principal reason probably lies in the fact that the exhaust gases of the normal two-cycle engine carry a considerable amount of scavenging air and are, therefore, of much lower temperature. Furthermore, the pressure at the opening of the exhaust ports is usually lower and the exhaust header of the engine is normally of much greater volume in order to reduce back pressure or resistance to scavenging. These conditions will be changed in the improved two-cycle engine of the future.

Now that the announcement of regular production of jet-propelled planes is made, it may have occurred to you that our aircraft friends have stolen a march on us, who are confined to the sea level regions, as far as the gas turbine is concerned. I do not think that is necessarily so—they merely took advantage of the simple but very important phenomenon that atmospheric pressure decreases with distance from sea level, something they have been successfully utilizing in turbocharging aircraft engines for many years. This gives them lower back-pressure and greater

pressure drop for operation of their gas turbines the higher they rise—therein may lie the solution of stratosphere flying at really high speeds. I have already referred to the ingenious device by which Buchi achieved the same effect by cylinder grouping and suggested that we improve it further by providing each cylinder with its own exhaust passage. When this suggestion is carried out for the two-cycle opposed-piston engine and the gas pressure at opening of exhaust ports is raised, there will be ample energy in the "ballistic" or Kadenacy effect of the escaping gases to drive a scavenging blower and thereby achieve a similar pressure drop as in aircraft. Eventually, this blower may be transformed into a compressor of higher pressure and the engine, turbine and compressor geared into one common unit as Sulzer Brothers is doing on their new engine. By going one more step, the engine output may be reduced to the requirements of the compressor and these two form a new type of unit—a power gas generator providing its exhaust gases mixed with surplus air for the turbine driving the propeller of a ship either through gears or through electric transmission. A number of experimental installations of this type for ship propulsion were made before the war by Gotaverken in Sweden.

The final and most attractive step in these series of developments is the elimination of the crank mechanism in the engine-compressor or gas generator unit. The resulting free-piston type of gas generator is being developed by Pescara interests in France and England. This last development will require considerable time and effort before a thoroughly practical means of ship propulsion will emerge. However, the Bureau is actively supporting it in this country, as well as the parallel of development of continuous-combustion type, where one turbine is driving an axial compressor and another one the useful load. Which of the two will succeed first or whether both methods will yield attractive solutions and be coexistent in the future is not given to us to know positively at this time but there is a lot of metallurgical progress needed before the fuel economy of the continuous-combustion type can be made good enough to consider seriously its adaptation in a Naval vessel, where economy of fuel is of greater importance than the reduced first cost.

I hope you have enjoyed this trip into the future and a preview of what is in store for the Diesel engine as far as I can see it.

I conclude with the admonition that it will take a lot of sweat and consecrated loyalty to our country and to the job at hand before worthwhile results will be forthcoming. Let us all pull together and let no one feel that his cooperation is not needed and his talents are too small to matter. Good luck!

APPROVAL TESTING OF NAVAL DEAERATING FEEDWATER HEATERS.

By Robert C. Adams.*

The advent of integral economizers in Naval boilers made necessary the provision of deaerating heaters for protection against corrosion. The thermal saving of the economizers is such that their semi-annual renewal would be less expensive than their elimination, but such frequent docking of a military vessel would severely limit its usefulness. Alternative methods for removal of dissolved oxygen, such as deactivation or chemical reduction, are prohibited by space and weight limitations. The need for deaerators having been established, the next requirement was a means for testing manufacturers' designs to insure a reasonable standard of performance after installation. The method of testing and evaluation is discussed in this paper.

In shop-testing of deaerators it is desirable to simulate the whole range of operating conditions which are to be expected in service. Most of these variations pertain to the feedwater entering the deaerator and consist of rate of flow, pressure, temperature and dissolved-oxygen content. The probable range in steam temperature and pressure also should be explored. Since Naval deaerators do not remain in a fixed position it would be desirable to investigate their performance at various angles from the vertical, but this ultimate simulation would require an elaborate test mounting on a rocking platform and is not attainable within reasonable cost.

A complete test set-up consists of the deaerator to be tested, an adequate supply of both feedwater and steam, and means for removal of the deaerated water. Rates of flow of feedwater, steam and effluent may be measured but it has been found equally satisfactory to measure either feedwater or effluent rate of flow, usually the former because its temperature is more constant, and to calculate the other two from thermal data, although this in-

* U. S. Naval Engineering Experiment Station, Annapolis, Md.

volves the assumption of no thermal loss from the deaerator shell. Pressure and temperature of inlet feedwater and steam and of effluent deaerated water and vented steam also are measured. An external condenser of ample size is provided so that the vented steam can be condensed and then weighed or measured. The pressure in the deaerator body below the spray head is considered as the operating pressure of the unit. Manometers are installed between this point and the steam and feedwater lines immediately outside the shell to measure the drop in pressure for both fluids in passage through the mixing devices and baffles. Figure 1 is the flow sheet of a complete test set-up.

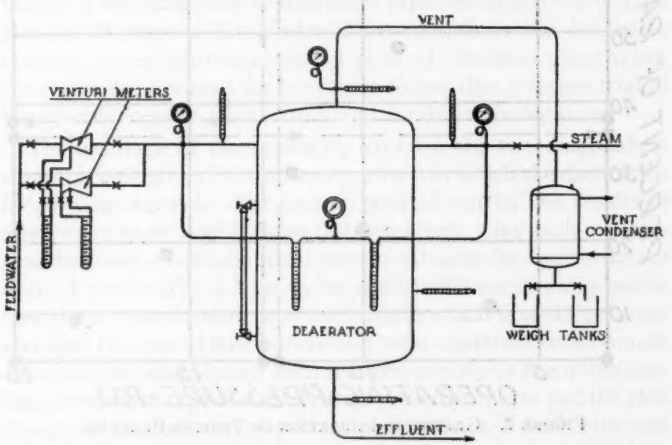

FIGURE 1. FLOW SHEET OF DEAERATOR—TEST ARRANGEMENT.

It might be expected that the principal variables affecting the completeness of deaeration would be the temperature and dissolved-oxygen content of the feedwater, the operating pressure, the proportion of vented steam, and the rate of flow. Investigation of the independent effects of five variables is a time-consuming and expensive task which we have attempted only once. Even then the exploration was not completed because the results indicated that with the usual fixed orifice in the vent line only operating pressure and rate of flow exerted a significant effect upon the dissolved-oxygen content of the deaerator effluent.

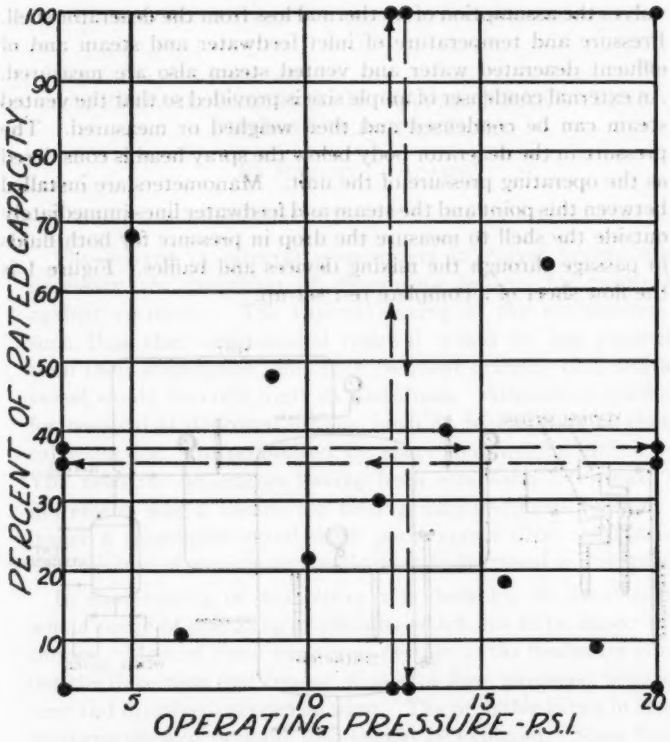

FIGURE 2. GRAPHICAL INDICATION OF TESTING PROGRAM.

The program of test runs for a deaerator appraisal accordingly is laid out graphically on pressure-rate axes. Such a lay-out is illustrated in Figure 2. Test runs are made at three of the four corners of the rectangle, representing the operating range of the deaerator. Since it is improbable that any unit will be operated at minimum pressure and maximum rate, the fourth corner is omitted. A sufficient number of runs are distributed along curvilinear diagonals to provide a reasonable representation of performance. These diagonals intersect at approximately 30 per cent of the full load of the unit, both because the poorest deaeration usually is encountered below half-load and because the majority of service operation is at low loads. These runs are

distributed so that no two are made at the same pressure or rate of flow. Purchase contracts usually require an evaluation of the performance at loads above the maximum rating and these exploratory runs usually are made at a median operating pressure. If testing facilities permit, these are extended to twice the rated capacity for determination of the ability of a single unit to provide feedwater for two engine-rooms in case of an emergency. With the exception of the single run at minimum load and pressure, the only restrictions on feedwater are that it shall be no warmer than 80 degrees F. and that it contain at least 3 ppm. of dissolved oxygen. In the exceptional run, the feedwater is maintained at the maximum temperature expected in service, usually about 140 degrees F. Under these conditions the feedwater scarcely can contain more than 1 ppm. of dissolved oxygen, but the steam requirement for heating is so low that a severe trial of the effectiveness of atomization and scrubbing is obtained.

The coverage of the operating area by the two diagonals is complete enough to indicate any portions in which the deaeration is less than normal. The area so pointed out by the results of any one or more runs is more fully explored. One such exploration disclosed a semi-elliptical area of sub-standard performance only 13 per cent by 1.2 psi. lying against the minimum-pressure boundary. In addition, runs are made in which first the pressure and then the rate of flow is increased from minimum to maximum as rapidly as possible and then reduced rapidly to the minimum. The pressure change is made at the most critical rate and the rate change at the pressure of poorest performance, if such rate and pressure have been found, otherwise at convenient, median values.

The analytical station is set up immediately adjacent to the heater. Care in arrangement of this station so that the work table is of convenient height, lighting is adequate, and an orderly flow of samples can be maintained without lost motion is an economical precaution since the hourly cost of conducting such tests is considerable. The method of analysis has been described in detail elsewhere[1]. An analytical team of three has been found most efficient: one to collect and fix samples; another to titrate them; and the third to record and calculate the results.

(1) Proceedings American Society for Testing Materials, Volume 43 (preprint 90, 1943 Annual Meeting).

The order of tests is agreed upon in advance between the analytical team and the operating crew, usually in accordance with concurrent demands on the power plant at the test location. The operating crew establishes the predetermined conditions and, upon signal, begins recording all operating pressures and temperatures at five-minute intervals. Simultaneously the analytical team begins collection and analysis of alternate effluent and feedwater samples until three effluent and two feedwater samples yield results in satisfactory agreement. If these results indicate a trend to lower or higher values the run is continued until the results become constant. As soon as reliable and satisfactory results have been obtained, the run is terminated, and the crew proceeds to establish the conditions for the next.

The runs under conditions of changing rate or pressure are primarily for testing the mechanical performance of the unit since the dissolved-oxygen determinations made on samples taken during these runs are of limited significance. The water level in the storage space of the unit is lowered to the minimum controllable in order to reduce the lag in change of the dissolved-oxygen content of the effluent following a change in effectiveness of deaeration. However the change from minimum to maximum, or the reverse, in either pressure or rate can be made in from thirty seconds to three minutes and about one minute is required for a single change of water in the sampling flasks. As a result only gross changes in dissolved-oxygen content can be detected. Here is a place where an electronic, continuous, dissolved-oxygen indicator would be valuable.

The voluminous data collected from twenty or more such runs require critical review to assure an accurate appraisal of the performance of the deaerator. Operating data from each run for periods before and after that in which the consistent analytical results were obtained are discarded. The remaining data, except for venturimanometer readings, are averaged and calibration corrections for the several gauges and thermometers are applied to the averaged results. The feedwater rate at each interval is calculated from the manometer readings using a simplified form of the A. S. M. E. formula:[2]

$$F = K \sqrt{hd}$$

(2) Mechanical Engineering Vol. 52, p. 968 (1930) or Chemical Engineers Handbook, Firs Edition, p. 699 (1934).

in which h = the manometer reading in inches of mercury, d = the specific gravity of the feedwater at the measured temperature, and K = the calibrated or calculated constant for the meter used. The individual feedwater rates are averaged.

The effluent rate is calculated from steam-table data using the formula:

$$D = \frac{F(H_1 - L_1) - V(H_1 - H_2)}{H_1 - L_2}.$$

The significance of the symbols is shown below:

D = deaerated discharge, pounds per hour.
F = feedwater, pounds per hour.
V = vented steam, pounds per hour.
H_1 = total heat of heating steam, Btu. per pound.
H_2 = total heat of vented steam, Btu. per pound.
L_1 = heat of feedwater, Btu. per pound.
L_2 = heat of deaerated discharge, Btu. per pound.
S = heating steam, pounds per hour.

The above formula is derived by simultaneous solution of the two verities:

$$SH_1 + FL_1 = VH_2 + DL_2; \text{ and } S + F = V + D.$$

Having calculated the discharge rate, the efficiency of the unit can be determined by substitution in the equation following:

$$\text{Efficiency} = \frac{D(L_2 - L_1)}{V(H_2 - L_1) + D(L_2 - L_1)}$$

Except during operation at minimum rate and pressure with hot feedwater the efficiency of a good deaerator will consistently exceed 0.98 and at high rates will approach 0.995. Even under the least favorable conditions cited above the efficiency usually is at least 0.8 although values as low as 0.6 have been encountered in trials.

The limited significance of the efficiency as calculated above should be recognized. It serves only as a crude, inverse index of the proportion of steam vented to remove the non-condensable

gases. Such a device must be provided for consideration in the evaluation of heater performance. Although vent loss is more closely related to shell pressure than to water rate the efficiency provides a more familiar and recognizable measure than alternate expressions which have been devised. Efficiency has no practical, thermal significance even though it is calculated from thermal data.

A final critical review of the individual dissolved-oxygen determinations is made. In spite of every effort to maintain the highest attainable accuracy there is an ever-present possibility of error. Consistent values, of whatever magnitude, are presumed to be correct, but cases where successive values are of varying magnitude raise the question as to whether the variation is in the accuracy of the determinations or in the performance of the deaerators. Such variations are subjected to appraisal by the differential equation:

$$\frac{dx}{dt} = \frac{D}{M}(A - x);$$ in which

x = dissolved oxygen content of the stored water and discharge in the deaerator at any instant

t = time, minutes

M = weight of stored water

A = dissolved oxygen content of water entering the storage space from the deaerating section.

Integration of the above and evaluation of the constant of integration yields:

$$A = \frac{I_2 e^{\frac{dt}{M}} - I_1}{e^{\frac{dt}{M}} - 1},$$

in which I_1 and I_2 are the successive dissolved-oxygen contents of the effluent as determined, and t is the time between collection of the two. If A is a reasonable value it is assumed that the deaerator is at fault. This is admittedly an arbitrary procedure, since erroneous determinations may still yield reasonable results, but the alternative is expensive, extended testing under fixed conditions. Fortunately, it has not had to be invoked frequently,

although negative values for A in one test led to final corrections in the method of analysis.

The method of testing and evaluation is believed to be rigorous but reasonable. The time and expense required for testing are held to the minimum for such a comprehensive test as is desirable for a prototype. The writer has enjoyed the development of the procedure and acknowledges the manifold assistance afforded by the several manufacturers. The development at many stages appeared to be a race between the manufacturer, endeavoring to build deaerators so nearly perfect that the dissolved-oxygen content of the effluent could not be determined with accuracy, and the analytical crew, trying to perfect such a method that no smallest trace of dissolved oxygen could escape detection. If it was a race, the competitors seem to have run a dead heat.

CABLE CALCULATIONS FOR SHIPBOARD POWER AND LIGHTING SYSTEMS.

By B. F. Greenberg.*

The cable calculations required in the design of shipboard lighting and power systems can be generally classified as follows:

(*a*) The determation of the voltage drop for a given load and cable size.

(*b*) The determination of the cable size in any section when the overall voltage drop in the system is the limiting factor.

The former calculations are characteristic of those required in the design of power systems; where, because of the relatively short cable runs on shipboard installations, current carrying capacity is usually the governing factor in the choice of cable size. Nevertheless, voltage drop calculations need to be made, particularly under motor starting conditions, to assure the satisfactory performance of the system.

On shipboard lighting systems the second type of calculation is most common. The efficiency and life of incandescent lamps, which constitute the largest proportion of the load on the lighting system, are greatly affected by the applied voltage; consequently, it is essential to maintain a uniform voltage level on the lighting system. This fact, combined with the relatively low voltage used on lighting systems, results in voltage drop being the limiting factor in the choice of cable size for almost all sections of a lighting system.

Part I—Cable Constants.

Pre-requisites for all power and lighting system cable calculations are the cable constants, i. e., the resistance and reactance. These constants are dependent upon the physical dimensions, construction and the assumed operating conditions of the cable.

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The resistance and reactance of standard Navy type THFA three conductor, 600 volt cables are given in Table I.

TABLE I.

"THFA" Cable Characteristics.

Cable		Resistance Per Conductor	Reactance Per Conductor	Reactance Per Conductor	Impedence Per Conductor
Size	Circular Mils	at 65°C.[1]	Non-magnetic Armored	Steel[2] Armored	Steel Armor
		ohms/1000[1]	ohms/1000[1]	ohms/1000[1]	ohms/1000[1]
3	2828	4.67	.0503	.0533	4.67 ∠0°39′
4	4494	2.89	.0462	.0490	2.89 ∠0°58′
9	9030	1.44	.0414	.0440	1.44 ∠1°45′
14	14350	.903	.0396	.0422	.909 ∠2°40′
23	22820	.569	.0571	.0396	.570 ∠3°59′
30	30780	.423	.0366	.0391	.424 ∠5°17′
40	38950	.334	.0355	.0380	.336 ∠6°30′
50	49020	.265	.0343	.0378	.268 ∠8°07′
60	59940	.218	.0349	.0375	.221 ∠9°46′
75	75850	.173	.0347	.0376	.177 ∠12°17′
100	98820	.132	.0338	.0368	.137 ∠15°36′
125	125050	.105	.0328	.0360	.111 ∠18°57′
150	157380	.0840	.0329	.0363	.0915 ∠23°23′
200	198860	.0665	.0319	.0356	.0754 ∠28°14′
250	250710	.0530	.0313	.0354	.0637 ∠33°45′
300	296660	.0449	.0312	.0358	.0574 ∠38°46′
350	414020	.0384	.0312	.0362	.0526 ∠43°27′
400	414020	.0324	.0300	.0354	.0479 ∠47°31′

1 = For resistance at other temperatures use formula

$$R_t = R_{65} \frac{(234.5 + t)}{(234.5 + 65)}.$$

2 = Based on 85 per cent coverage.

CABLE RESISTANCE.

The resistance of a copper conductor depends upon the temperature, stranding, spiralling and tinning of the conductor and the frequency of the current flowing. Considering the factors of stranding, spiralling and tinning, a typical value of resistivity at 30 degrees C. is 11.5 ohms per circular mil per foot, compared to a value of 10.8 ohms for a solid copper conductor. The temperature at which a conductor operates depends upon the ambient temperature, and the current carried by the conductor. Standard Navy cable carrying rated current in an ambient temperature of 50 degrees C. reaches a final copper temperature of 105 degrees C. However, for an average ambient temperature of 40 degrees C. and under motor starting conditions, the average value of conductor temperature may be assumed equal to 65 degrees C.

The increase in resistance, due to the skin effect when alternating current flows in the conductor, is negligible for conductor sizes of 400,000 circular mils and a frequency of 60 cycles. Similarly, the increase in resistance to alternating current due to the proximity effect of steel armor or other adjacent metal may be neglected. Accordingly, the values of resistance given in Table I are the same for both alternating and direct currents.

CABLE REACTANCE.

The reactance per phase of a three conductor cable with non-magnetic armor, at 60 cycles, can be calculated from the formula:

$$X = .0529 \log_{10} \frac{S}{.752\ r} \qquad (1)$$

where:

X = cable reactance in ohms per phase per 1000 feet at 60 cycles

S = conductor spacing, i. e., overall conductor diameter

r = outside radius of conductor

Note: Values of S and r are obtained from cable construction data.

When the cable armor is of magnetic material, i. e., steel, the flux linkages per ampere in each conductor are increased because

of the increased permeability of the magnetic path. Therefore the reactance of a cable with magnetic armor is greater than that of a cable with non-magnetic armor. Moreover, a similar increase in the reactance of cable is produced by the presence of adjacent steel plates. In all cases where magnetic material is present, the reactance is a function of the magnitude of the current flowing because of saturation. This effect is, in general, not appreciable, except at very high currents; usually many times the rated current of the cable. Under this condition the reactance of steel armored cable is equal to that of the cable with non-magnetic armor.

The increase in the reactance of a three conductor cable due to magnetic material surrounding or adjacent to it is not readily calculable. The values given in Table I are based on tests, and show that the reactance of size THFA-400 steel armored standard Navy cable with 85 per cent coverage is 18 per cent greater than that of cable with non-magnetic armor. On shipboard installations of non-magnetic armored cable the close proximity of steel bulkheads will cause the reactance to be approximately equal to that of steel armored cable. On the other hand, steel armored cable, being partially shielded magnetically will be relatively unaffected by adjacent steel plates.

Part II—Power System Cable Calculations.

On an AC power system, the exact value of the per cent voltage drop in a cable for any set of conditions can be calculated from the equation:

$$\%\,\text{V. D.} = \frac{E - V}{E} = 100\left[1 + \frac{Il}{E}z\cos(\theta - \beta) - \sqrt{1 - \left(\frac{Il}{E}\right)^2 z^2 \sin^2(\theta - \beta)}\right] \qquad (2)$$

where:

$\%$ V. D. = $\%$ voltage drop
E = line voltage
V = terminal voltage
I = current
l = length of conductor
z = cable impedance per foot
$\theta = \cos^{-1}$ (power factor)
$\beta = \tan^{-1}\left(\frac{X}{r}\right)$ of the cable

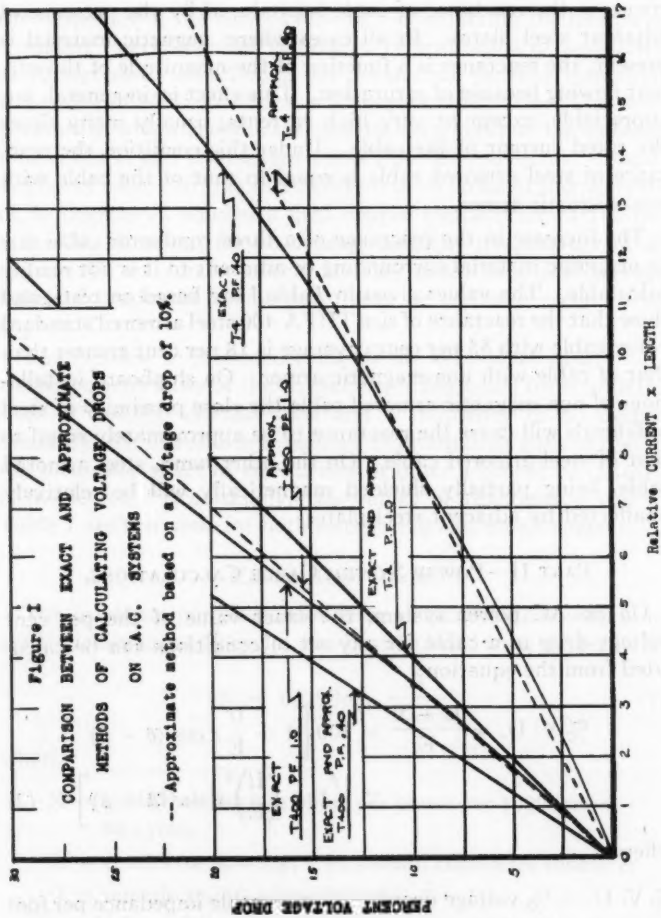


Figure I
COMPARISON BETWEEN EXACT AND APPROXIMATE METHODS OF CALCULATING VOLTAGE DROPS ON A.C. SYSTEMS
-- Approximate method based on a voltage drop of 10%.
EXACT T-4 P.F. .80
APPROX. T-4 P.F. .80
APPROX. T-4 P.F. 1.0
EXACT T-4 P.F. 1.0
EXACT AND APPROX. T-4 P.F. 1.0
EXACT AND APPROX. T-4 P.F. .80
PERCENT VOLTAGE DROP
Relative CURRENT x LENGTH

From the above equation it is apparent that there is no single factor which, when multiplied by the current and length, would give the exact voltage drop. As a result, this equation is cumbersome for practical applications. For this reason, a simplified formula has been derived, (appendix A,) which can be applied for the range of voltage drops usually encountered.

The simplified equation for per cent voltage drop is equal to:

$$\% \text{ V. D.} = I/F \tag{3}$$

where

F = "voltage drop factor"

$$= \frac{z \cos (\theta - \beta)}{E}\left[1 + \frac{d_0 \tan^2 (\theta - \beta)}{200}\right]$$

d_0 = per cent voltage drop at which the factor directly applies.

The voltage drop factors given in Table II were obtained by substituting in equation (3) the cable characteristics, Table I, and an anticipated voltage drop of 10 per cent. The difference between the exact values, equation (2) and the approximate solution, equation (3) is given in Figure I for different cable sizes and voltage drops. From Figure I it is evident that for values of voltage drops less than 10 per cent the approximate solution yields values slightly below the exact values; while, for values above 10 per cent, the exact value is greater than the approximate value. The maximum accuracy for the approximate method was obtained for values in the order of 10 per cent, as, in most cases, voltage drop calculations are made so as to limit the voltage drop to 10 per cent under motor starting conditions.

For DC systems, the "voltage drop factors" as given in Table II were obtained from the formula:

$$\% \text{ V. D.} = \frac{IR \times 2l \times 1.00}{E} = I/F \tag{4}$$

where

l = single length of conductor (feet)

R = resistance per foot.

The "voltage drop factors" given in columns 3–12 of Table II can be applied directly to obtain the voltage drop on a 450 volt,

TABLE II—*A. C. and D. C. Voltage Drop Factors Power Systems*

Cable Size	Cable Circular Mils	Voltage Drop Factors × 10^{-5} A. C., 450 Volt, Three Phase, 60 Cycle Power Factor 1.0	.95	.90	.85	.80	.70	.60	.50	.40	.30	D. C. 120v	120v	Cable Size
3	2828	180	172	164	157	149	133	119	105	91	81	765	384	3
4	4494	111	107	102	97	92	83	74	65	57	51	481	240	4
9	9030	55	53	51	49	46	42	37	33	29	26	240	119	9
14	14350	35	33	32	31	30	27	24	21	19	16	151	76	14
23	22820	22	21	20	20	19	17	15	13	12	11	95	48	23
30	30780	16	16	15	15	14	13	12	10	9.2	8.2	71	35	30
40	38980	13	13	12	12	11	10	9.4	8.4	7.4	6.6	56	28	40
50	49020	10	10	9.8	9.5	9.2	8.4	7.7	6.9	6.1	5.4	44	22	50
60	59940	8.4	8.4	8.2	8.0	7.7	7.1	6.4	5.9	5.2	4.6	36	18	60
75	75840	6.7	6.8	6.6	6.4	6.3	5.8	5.3	4.9	4.4	3.9	29	14	75
100	98820	5.1	5.3	5.2	5.1	5.0	4.7	4.3	3.9	3.6	3.2	22	11	100
125	125050	4.1	4.3	4.2	4.2	4.1	3.9	3.6	3.3	3.1	2.6	17	8.7	125
150	157380	3.3	3.5	3.5	3.5	3.4	3.3	3.1	2.9	2.7	2.5	14	6.9	150
200	198860	2.6	2.9	2.9	2.9	2.9	2.8	2.7	2.5	2.3	2.2	11	5.5	200
250	250710	2.1	2.4	2.4	2.4	2.4	2.4	2.3	2.2	2.1	2.0	8.7	4.3	250
300	296660	1.8	2.1	2.1	2.2	2.2	2.2	2.1	2.1	2.0	1.9	7.3	3.7	300
350	349804	1.5	1.9	1.9	2.0	2.0	2.0	2.0	1.9	1.9	1.8	—	—	350
400	414020	1.3	1.6	1.7	1.8	1.8	1.8	1.8	1.8	1.7	1.7	5.3	2.6	400

NOTES

1—For A. C. Systems Multiply "Voltage Drop Factors" by—

Voltage	Single Phase	Three Phase
120	4.34	3.75
240	2.17	1.88
450	1.15	1.00

2—Drop factors are based on cable constants given in Table I.

three phase, 60 cycle system. For any other type AC system, the "voltage drop factors" should be multiplied by the constants corresponding to the type of system used. The "voltage drop factors" for DC systems are given in columns 13–14 of Table II.

Examples of Power System Cable Calculations.

The method of calculating voltage drops in power systems can be illustrated by the following examples:

(*a*) Find the per cent voltage drop on a 450 volt, three phase system. The load power factor is .40, the current is 300 amperes, length 250 feet, and cable size THFA-23.

Solution: From Table II the voltage drop factor for THFA-23, at 40 per cent power factor is 12×10^{-5}. Thus, the per cent voltage drop is equal to per cent V. D. = $I/F = 300 \times 250 \times 12 \times 10^{-5} = 9$ per cent.

(*b*) Find the per cent voltage drop on a 240 volt DC system. The cable size is DHFA-30, current 85 amperes and length 130 feet.

Solution: From Table II the voltage drop factor for DHFA cable at 240 volts is equal to 33×10^{-5}. The per cent voltage drop is equal to: per cent V. D. = $I/F = 85 \times 130 \times 35 \times 10^{-5} = 3.9$ per cent.

Part III—Method of Choosing Optimum Cable Sizes for Lighting Systems.

The choice of cable sizes in the design of lighting systems is, in most cases, limited by voltage drop. The cable sizes in each section must be chosen so that the sum of the voltage drops from the beginning to the end of the system does not exceed a specified value. The optimum cable size in any section is then that size which yields the minimum weight of cable with no section exceeding rated current and with the overall per cent voltage drop equal to but not greater than the specified value.

A method of calculating lighting system cable sizes is presented herein whereby the entire system is considered as a whole, and cable size in each section can be calculated by a systematic procedure. This procedure eliminates trial and error methods and results in a saving of both time and material.

2—Drop factors are based on cable constants given in Table I.

Fig. 2 a - DIAGRAM OF SIMPLIFIED LIGHTING DISTRIBUTION SYSTEM ---- Equal Sections

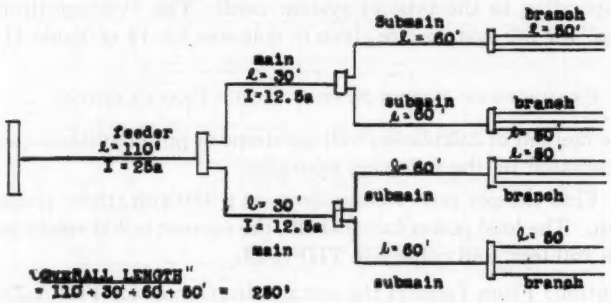
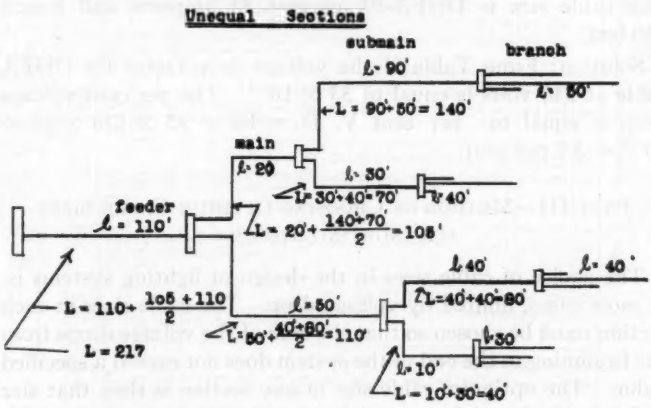


Fig. 2 b - LIGHTING DISTRIBUTION SYSTEM

Unequal Sections



General Principles.

The arrangement of cables in a typical lighting system is shown in Figure 2. As shown in Appendix B the determination of the optimum cable size is based on the principle that the voltage drop per foot in any part of the lighting circuit, regardless of the current carried in that section should be equal to a constant. This constant depends on the average overall length

of the system and the allowable overall voltage drop. Thus the voltage drop allocated to a particular section is to the maximum allowable drop in the system as the length of the section is to the average combined length of the feeder, main, submain, and branch circuits. For example, refer to Figure 2a wherein the length of each corresponding section (main, submain and branch) is equal. In this case the average overall length is equal to the actual overall length or 250 feet. Therefore, if the allowable overall voltage drop is 6 per cent, the voltage drop allocated to the feeder should be $\frac{110}{250}$(6 per cent) = 2.65 per cent and the drop allocated to the main should be $\frac{30}{250}$ (6 per cent) = .72 per cent.

Expressed mathematically, the voltage drop allocated to each section in accordance with the above principle is given by:

$$\frac{d}{l} = \frac{D}{L} = \text{constant} \qquad (5)$$

In this formula, the per cent voltage drop, d, in any section, based on an average conductor temperature of 45 degrees C. and a system voltage of 120 volts is equal to:

$$d = \frac{20.0\ Il}{C.M.} \text{ for single phase and DC systems} \qquad (6a)$$

$$d = \frac{17.3\ Il}{C.M.} \text{ for three phase systems} \qquad (6b)$$

where:

I = resultant current in conductor under consideration
C. M. = cross sectional area of the conductor (circular mils)

Combining equations 5 and 6, the optimum cable size in any section can be obtained directly from the formula:

$$C.M. = \frac{20.0\ I\ L}{D} \qquad (7a)$$

$$C.M. = \frac{17.3\ I\ L}{D} \qquad (7b)$$

For example, with an allowable overall per cent voltage drop of 6 per cent, a 120 volt DC system, a current of 25 amperes, the optimum cable size in the feeder, Figure 2 is equal to:

$$\text{C. M.} = \frac{20 \times 25 \times 250}{6} = 20{,}800 \text{ circular mils.}$$

The cable size in the mains, with a current of 12.5 amperes is equal to:

$$\text{C. M.} = \frac{20 \times 12.5 \times 250}{6} = 10{,}400 \text{ circular mils.}$$

In an actual lighting distribution system where the lengths of corresponding sections (mains, submains, and branches) are not equal, the average overall length of the system when viewed from each main or submain will, in general, be different. Furthermore, the available cable sizes are limited to a finite number. In order to take these factors into account, the method of calculating cable sizes as outlined above must be modified.

By referring to Figure 2b it will be evident that the average overall length of the system as viewed from a submain is equal to the length of the submain plus the average length of the branch circuits connected to it. Similarly, the average overall length of the system as viewed from a main is equal to the length of the main plus the sum of the average overall lengths of the system as viewed from each submain connected to it divided by the number of submains. By determining thus the average overall length of the system for each section, the general principle set forth in equation 5 can be applied by successively eliminating each section of the system from the calculations once the cable size and voltage drop in each section is calculated. For example, the cable size in a feeder is determined by substituting for "L" in equation 7, the average overall length of the system as viewed from the feeder, and for "D" the allowable overall voltage. The cable size for the main is determined in a similar manner by substituting for "L" the average overall length of the system as viewed from the main and for "D" the allowable overall voltage drop minus the voltage drop in the feeder.

In making the calculation of the average overall length, if the length of any section is at a voltage other than the reference voltage then the length of that section should be multiplied by

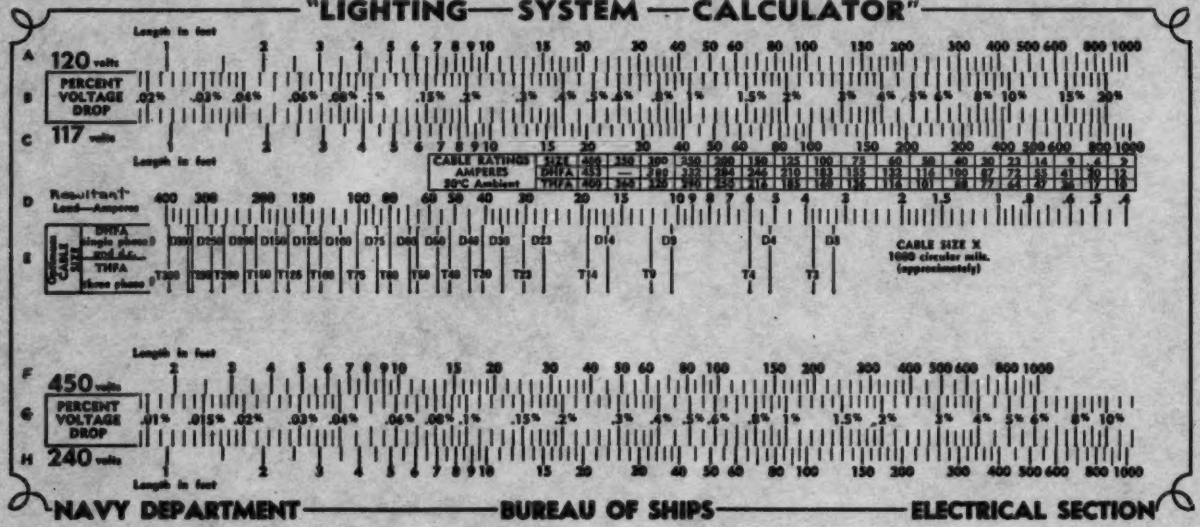

"LIGHTING—SYSTEM—CALCULATOR"
Length in feet
120 volts
PERCENT VOLTAGE DROP
117 volts
CABLE RATINGS AMPERES 50°C Ambient
Resultant Load—Amperes
CABLE SIZE
DMFA single phase 3 wire d.c.
TMFA three phase
CABLE SIZE X 1000 circular mils. (approximately)
450 volts
240 volts
NAVY DEPARTMENT—BUREAU OF SHIPS—ELECTRICAL SECTION

FIGURE 3.

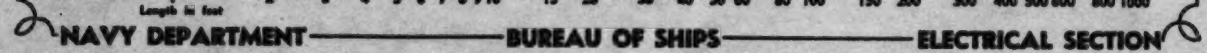


FIGURE 3.

LIGHTING SYSTEM CALCULATOR

For all types of lighting systems the "Calculator" provides a means for the rapid determination of optimum cable size and voltage drop, which can be used in conjunction with Bureau drawing 9-S-5474-L.

INSTRUCTION.

A. *Optimum Cable Size.*

1. To find the optimum cable size in any section of a lighting system, set the allowable overall per cent voltage drop (Scale B) opposite the overall length of the system (Scale A or C).
2. Read the cable size (Scale E) opposite the resultant current (Scale D). For single phase and d-c systems use the next largest DHFA cable size, and for three phase systems use the next largest THFA cable size.

B. *Voltage Drop.*

1. To find the per cent voltage drop in any section, set the cable size (Scale E) opposite the resultant current (Scale D).
2. Read the per cent voltage drop opposite the length of the section on the scale which corresponds to the voltage of the conductor (Scale A, C, F, or H).

EXAMPLE.

A. Find the optimum feeder size for a three phase, 120 volt system with an average overall allowable per cent voltage drop of 6 per cent. The resultant current in the feeder is 100 amperes.
Set 6 per cent (Scale B) opposite 260 feet (Scale A), then opposite the resultant current of 100 amperes (Scale D), the next largest cable size is THFA 75 (Scale E).

B. Find the per cent voltage drop in the feeder of the above system, using the resultant current of 100 amperes and a feeder length of 50 feet. Set the cable size T-75 (Scale E) opposite the resultant current of 100 amperes (Scale D); then the per cent voltage drop (Scale B), opposite the length of 50 feet (Scale A), is 1.1 per cent. If the feeder voltage is 450 volts, then the voltage drop (Scale G) opposite the feeder length (Scale F), is .3 per cent.

Note: The calculation of per cent voltage drop is based on the formula:

$$d = 12 \times 2l \times I \times 100/CM \times E \text{ (single phase and d-c systems)}$$

$$d = 12 \times 1.732\, I \times l \times 100/CM \times E \text{ (three phase system)}$$

Where resistivity of the cable = 12 ohm per circular mil foot at 45 degrees C.

l = length of single conductor

I = resultant load current

E = line voltage

the ratio of the reference voltage to the voltage in the section. For example, if the system is calculated on the basis of 120 volts then the length of any part of the system at 450 volts should be multiplied by $\left(\frac{120}{450}\right) = .27$ before including the length of that section in the overall length.

In order to simplify calculations of cable sizes and voltage drops on lighting systems the special slide rule shown in Figure 3 has been developed. The slide rule is based on the constants of standard Navy cable, and enables the determination of cable size and voltage drop to be made in a single operation.

Example of Cable Size Calculation on Typical Lighting System.

The following example, Figure 4, illustrates the procedure followed for calculating cable sizes, using Navy Standard Cable on a 120 volt three phase system. The allowable per cent voltage drop is 6 per cent.

1. From Figure 4 the combined length of submain and branch circuit should be calculated for one branch circuit connected to each submain. The branch circuit chosen should be the one with the greatest voltage drop. Thus, the combined length as viewed from the submains 1-F1A, 1-F1B, 1-F1C is equal to 100, 80, and 40 feet respectively.

2. Working back in the system the overall length from the beginning of the main to the end of the system is equal to the length of the main plus the average of the combined length of the submain and branch connected to the main. For example, the overall length from the main 1-F1 to the end of the system is equal to the length of the main 65 feet, plus the average of the combined length of submain and branch as viewed from the submain 1-F1A, L-F1B, 1-F1C. Thus $L = 65 + \frac{100 + 80 + 40}{3}$ = 140 feet.

Similarly, the overall length as viewed from the main 2-F1 and 3-F1 is equal to 110 feet and 120 feet respectively.

3. The overall length of the system viewed from the feeder can be calculated by adding the feeder length to the average overall length from the main to the end of the system. Thus, the

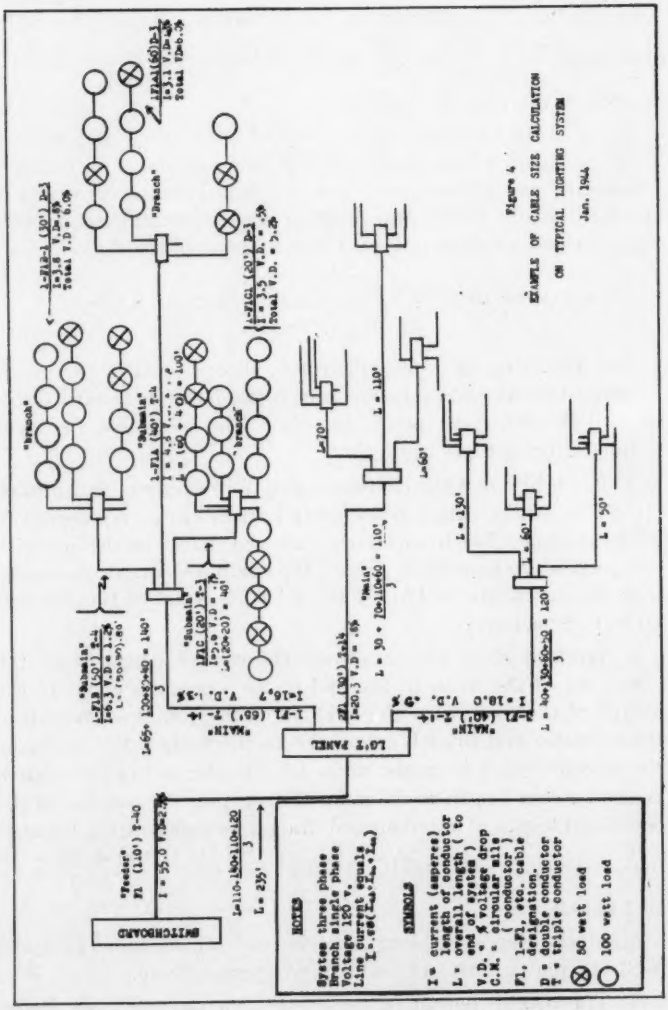


Figure 4.
EXAMPLE OF CABLE SIZE CALCULATION
ON TYPICAL LIGHTING SYSTEM
Jan. 1944

average overall length from the switchboard to the end of the system is equal to $L = 110 + \frac{140 + 110 + 120}{3} = 235$ feet.

The optimum cable size in the feeder can be calculated from equation 7b and is equal to:

$$C.\ M. = \frac{17.3 \times 55 \times 235}{6} = 37{,}000 \text{ circular mils.}$$

The next largest Navy standard cable size is T-40, (38,980 circular mils). The voltage drop resulting from the use of T-40 cable is 2.7 per cent.

4. The optimum cable size of the main 1-F1 can be calculated as follows: the allowable drop left in the system is 6.0 − 2.7 = 3.3 per cent. The average overall length from the main to the end of the system as viewed from 1-F1 is equal to 140 feet. The optimum cable size in the main 1-F1 is equal to:

$$C.\ M. = \frac{17.3 \times 16.7 \times 140}{3.3} = 12{,}200 \text{ circular mils.}$$

The installed Navy standard cable size is T-14 (14350) circular mils, and the resulting voltage drop is 1.3 per cent. The available voltage drop in the submain and branch circuit equals 3.3 − 13. = 2.0 per cent.

5. The cable size in the submain 1-F1A can be calculated from the formula $C.\ M. = \frac{17.3 \times 4.8 \times 100}{2.0} = 4150$ circular mils, the installed cable size T-4 (4494 circular mils), the voltage drop is .7 per cent. The voltage drop remaining in the circuit.

2.0 − .7 = 1.3 per cent is equal to the permissible drop in the branch circuit. The installed cable size in the branch circuit, 1F1A is D-3 (2828 circular mils), the voltage drop is 1.3 per cent. The overall voltage drop from the switchboard is 6 per cent.

6. The cable size in the submain 1-F1B is equal to $C.\ M. = \frac{17.3 \times 6.3 \times 80}{2.0} = 4360$ circular mils. The installed cable size is T-4 (4494 circular mils) and the resulting voltage drop 1.2 per cent. The allowable voltage drop in branch circuit, I-F1B1 is then equal to 2.0 − 1.2 = .8 per cent. The installed cable size is D-3 (2828 circular mils) and the voltage drop is .8 per cent.

A total per cent voltage drop from switchboard equals 6.0 per cent.

The method of calculating cable sizes as illustrated in the above example enables the optimum system design to be obtained with maximum utilization of cable and without the use of cut and try methods. It should be noted that no one set of rules could cover the many variations found in all lighting distribution systems and the ultimate usefulness of the method presented herein depends on the proper application of the principles to each particular problem.

The author wishes to thank Mr. P. N. Ross for his help in preparing this paper.

APPENDIX A

DERIVATION OF VOLTAGE DROP CONSTANTS.

$$d = 100\left(1 + \frac{IZ}{E}\cos\alpha - \sqrt{1 - \left(\frac{Il}{E}\right)^2 z^2 \sin^2\alpha}\right) \qquad (1)$$

where

α = per cent voltage drop

$$\alpha = \cos^{-1}(\text{P. F.}) - \cos^{-1}\left(\frac{r}{z}\right)$$

$$Z = lz$$

First approximation:

$$d = 100\left(\frac{Il}{E} z\cos\alpha + \left(\frac{Il}{E}\right)^2 \frac{z^2 \sin^2\alpha}{2}\right) \qquad (2)$$

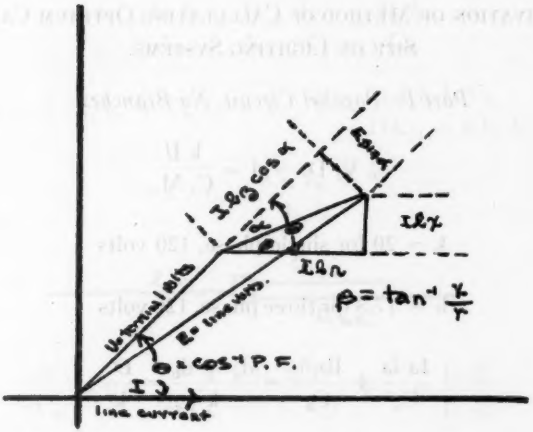
where

$$\left(1 - \frac{1}{2}\left(\frac{Il}{E}\right)^2 z^2 \sin^2\alpha\right) \cong 1 - \left(\frac{Il}{E}\right)^2 z^2 \sin^2\alpha + \frac{1}{4}\left(\frac{Il}{E}\right)^4 z^4 \sin^4\alpha$$

$$d = Il\,A + (Il)^2\,B \qquad (3)$$

$$A = \frac{z\cos\alpha\,(100)}{E}$$

$$B = \frac{z^2 \sin^2 \alpha \,(100)}{2\,E^2}$$



Second approximation:

$$d = Il\,F \tag{4}$$

$$F = A + Il\,B$$

since

$$A >> B$$

$$Il \propto \frac{d_0}{A}$$

$$F = A\left(1 + \frac{d_0 B}{A^2}\right)$$

$$F = \frac{z}{E}\cos\alpha\left(1 + \frac{d_0\, z^2 \sin^2 \alpha\,(100)}{2\, z^2 \cos^2 \alpha\,(100)^2}\right)$$

$$d = Il\left[\frac{z}{E}\cos\alpha\left(1 + \frac{d_0 \tan^2 \alpha}{200}\right)\right] \tag{5}$$

APPENDIX B.

DERIVATION OF METHOD OF CALCULATING OPTIMUM CABLE SIZE ON LIGHTING SYSTEMS.

Part I—Parallel Circuit, No Branches.

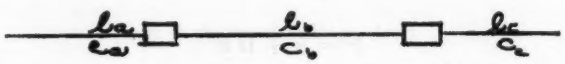
$$\% \text{ V. D.} = d = \frac{k\, Il}{C.\, M.} \tag{1}$$

$k = 20$ for single phase, 120 volts

$k = 17.3$ for three phase, 120 volts

$$\frac{Ia\, la}{C_a} + \frac{Ib\, lb}{C_b} = \frac{d_a + d_b}{k} = \frac{D}{k} \tag{2}$$

$$C_a\, l_a + C_b\, l_b = \text{Volume} = \rho W;\ V = C_a\, l_a + \frac{k\, I_b\, l_b^2\, C_a}{D\, C_a - k\, I_a\, l_a} \tag{3}$$



$$\frac{dV}{dC_a} = 0 = l_a + \frac{-\,k^2\, I_a^{\,2}\, l_b^{\,2}\, l_a}{(DC - k\, I_a\, l_a)^2} \tag{4}$$

$$C_a = \frac{k}{D}\sqrt{I_a}\left[l_a\sqrt{I_a} + l_b\sqrt{I_b}\right];$$

$$C_b = \frac{k}{D}\sqrt{I_b}\left[l_a\sqrt{I_a} + l_b\sqrt{I_b}\right] \tag{5}$$

$$C_x = \frac{k}{D}\sqrt{I_x}\left[l_a\sqrt{I_a} + l_b\sqrt{I_b} + l_c\sqrt{I_c} + \cdots l_x\sqrt{I_x}\right] \tag{6}$$

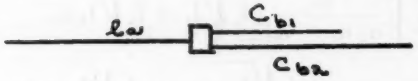
$$C_a = C_b\sqrt{\frac{I_a}{I_b}},\quad C_b = C_c\sqrt{\frac{I_b}{I_c}},\quad C_x = C_y\sqrt{\frac{I_x}{I_y}} \tag{7}$$

Part II—Multi-Branch Circuits, Unequal Lengths and Currents

$$\frac{I_a l_a}{C_a} + \frac{I_{b_1} l_{b_1}}{C_{b_1}} = \frac{D}{k} = \frac{I_a l_a}{C_a} + \frac{I_{b_2} l_{b_2}}{C_{b_2}} \tag{8}$$

$$C_a l_a + C_{b_1} l_{b_1} + C_{b_2} l_{b_2} = V; \quad V = \frac{k l_a I_a C_{b_1}}{D C_{b_1} - k I_{b_1} l_{b_1}}$$

$$+ C_{b_1} l_{b_1} + \frac{C_{b_1} I_{b_2} l_{b_2}^2}{I_{b_1} l_{b_1}} \tag{9}$$



$$\frac{dV}{dC_{b_1}} = 0 = \frac{-k^2 l_a^2 I_{b_1} l_{b_1} I_a}{(D C_{b_1} - k I_{b_1} l_{b_1})^2} + l_{b_1} + \frac{I_{b_2} l_{b_2}^2}{I_{b_1} l_{b_1}} \tag{10}$$

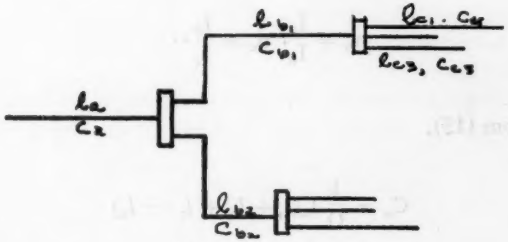
$$C_{b_1} = \frac{k}{D} \sqrt{\frac{I_{b_1}}{n_{b_1}}} \left[l_a \sqrt{I_a} + l_{b_1} \sqrt{I_{b_1} M_{b_1}} \right] \tag{11}$$

where

$$n_{b_1} = \frac{I_{b_1} l_{b_1}^2 + I_{b_2} l_{b_2}^2}{I_{b_1} l_{b_1}^2}$$

$$C_{b_2} = \frac{k}{D} \sqrt{\frac{I_{b_2}}{n_{b_2}}} \left[l_a \sqrt{I_a} + l_{b_2} \sqrt{I_{b_2} n_{b_2}} \right]$$

$$C_z = \frac{k}{D} \sqrt{I_a} \left[l_a \sqrt{I_a} + l_{b_2} \sqrt{I_{b_2} n_{b_2}} \right]$$



$$C_a = C_{b_2} \sqrt{\frac{I_a\, n_{b_2}}{I_{b_2}}} = C_{b_2} \sqrt{\frac{I_a\, n_{a_1}}{I_{b_1}}} \tag{12}$$

$$C_{b_1} = C_{c_1} \sqrt{\frac{I_{b_1}\, n_{c_1}}{I_{c_1}}}; \quad C_a = C_{c_1} \sqrt{\frac{I_a\, n_{c_1}\, n_{b_1}}{I_{c_1}}} \tag{13}$$

$$C_x = C_y \sqrt{\frac{I_x\, n_b\, n_c\, n_d \cdots n_x}{I_y\, n_b\, n_c\, n_d \cdots n_y}} \tag{14}$$

$$C_x = \frac{k}{D} \sqrt{\frac{I_x}{n_b\, n_c \cdots n_x}} \left[l_a \sqrt{I_a} + l_b \sqrt{I_b\, n_b} + l_c \sqrt{I_c\, n_b\, n_c} \cdots l_x \sqrt{I_y\, n_b\, n_c \cdots n_x} \right] \tag{15}$$

Part III—Multi-Branch Circuits, Equal Lengths

When lengths of respective sections are equal:

$$n_{b_1} = \frac{I_{b_1} + I_{b_2}}{I_{b_1}} \tag{16}$$

Assuming

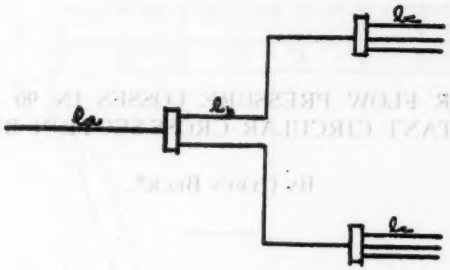
$$I_{b_1} = I_{b_2} \tag{17}$$

then

$$n_b = \frac{I_a}{I_b},\ n_c = \frac{I_b}{I_c} \cdots$$

∴ From (15),

$$C_a = \frac{k}{D} I_a (l_a + l_b + l_c \cdots l_x) \tag{18}$$



$$C_x = \frac{k}{D} I_x L \tag{19}$$

where

$$L = l_a + l_b + l_c \cdots l_x$$

$$d_x = \frac{k I_x l_x}{C_x} \tag{20}$$

$$\frac{d_x}{D} = \frac{l_x}{L} \tag{21}$$

LAMINAR FLOW PRESSURE LOSSES IN 90 DEGREE CONSTANT CIRCULAR CROSS-SECTION BENDS.

By Cyrus Beck*.

Note: The opinions or assertions contained in this article are the private ones of the writer and are not to be construed as official or reflecting the views of the Navy Department or the Naval service at large.

I—Introduction.

Original equivalent length data for 3½ inch I.P.S. valves, bends and fittings together with an examination of the character of the fluid resistance were presented in two previous articles. This paper, the third of a series of four, represents a study of pressure losses in 90 degree bends covering a Reynolds number range of 30 to 1000.

So far as is known, a comprehensive theory of fluid motion in circular bends has not yet been published. This is hardly surprising in view of the lack of agreement of the various experimental studies. "Indeed, almost as many contradictory curves have been published as there have been investigators engaged in such research."†

Curvilinear motion in 90 degree circular constant cross-section bends offers a particular challenge because the obvious geometrical regularity and symmetry would seem to indicate a comparatively direct and simple solution. Experimental results, however, show rather conclusively that despite the tremendous advances in fluid mechanics, curved flow must still be considered a complex phenomenon.

As defined previously, the equivalent length in diameters is the ratio of the pressure drop measured across a bend after sub-

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* Assistant Mechanical Engineer, Naval Boiler and Turbine Laboratory.

† Rouse, H., Fluid Mechanics for Hydraulic Engineers, McGraw-Hill Book Company New York, 1938. (P. 270.)

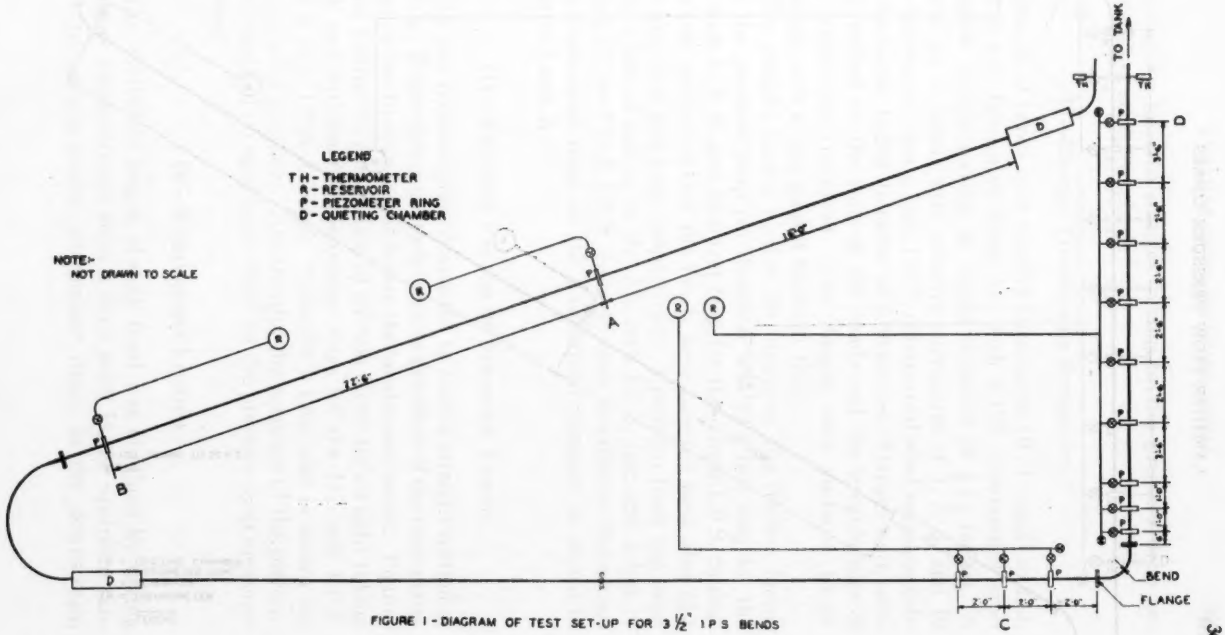

FIGURE 1 - DIAGRAM OF TEST SET-UP FOR 3 1/2" I P S BENDS

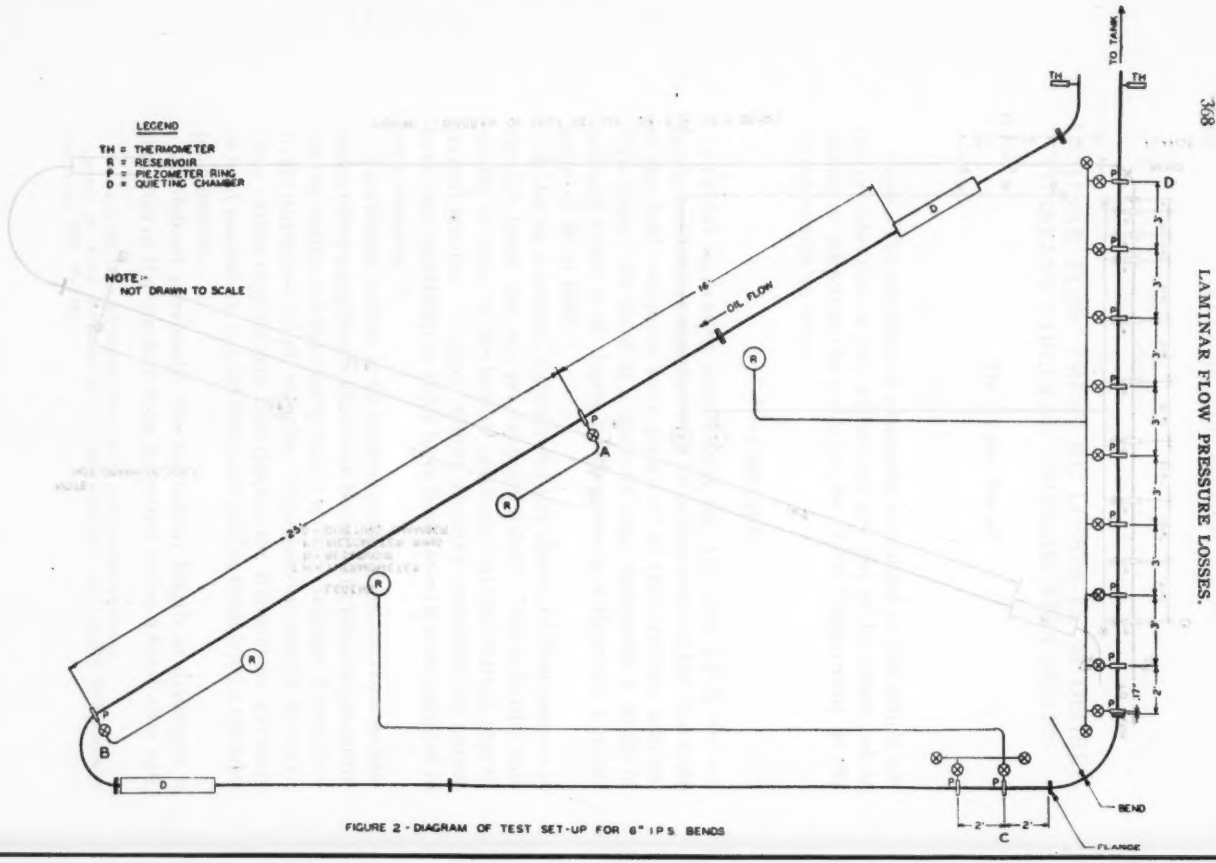

FIGURE 2 - DIAGRAM OF TEST SET-UP FOR 6" I.P.S. BENDS

tracting the loss for straight tubing to the pressure drop per diameter of straight tubing.

II—Bends, Tubing and Apparatus.

One set of bends with relative curvatures (R/d) equal to 2, 3, and 6 was fabricated from 3½ inch I.P.S. commercial steel seamless tubing having an inside diameter of 3.71 inches. A second set of bends with relative curvatures of 2, 3, 6, and 10 was fabricated from 6 inch I.P.S. commercial steel seamless tubing having an inside diameter of 6.36 inches. Flanges were carefully welded on the ends of the bends and the irregularities of the welds were removed. The flanges were surfaced to fit together with a thin gasket between them.

The straight tubing section for obtaining the friction factor and the pressure drop per diameter was 22.5 feet long for the 3½ inch I. P. S. and 25.0 feet long for the 6 inch I.P.S. tubing. The test sections where the bends were inserted were also 22.5 feet and 25.0 feet long, respectively. Upstream from the bend were 4 feet of tubing in the 3½ inch I.P.S. line and 2 feet of tubing in the 6 inch I.P.S. line. In each installation there were nine piezometer rings on the downstream tangent as shown in Figures 1 and 2.

III—Friction Factor of Straight Tubing.

The two tangents were connected to form a straight section of tubing. Runs were made to determine the effect of the piezometer rings on the friction factor before the bends were tested. Figures 3 and 4 show the comparison of friction factor for straight tubing with and without the piezometer rings for the 3½ inch I.P.S. and 6 inch I.P.S. tubings. Poiseuille's Law also is shown for comparative purposes. Apparently, the presence of the piezometer rings had no significant effect on the pressure drop measurements.

IV—Equivalent Length.

The equivalent length of each bend was obtained by taking pressure measurements from a fixed point in the upstream tangent to various points (piezometer rings) in the downstream

FIGURE 2 - DIAGRAM OF TEST SET-UP FOR 6" I.P.S. BENDS

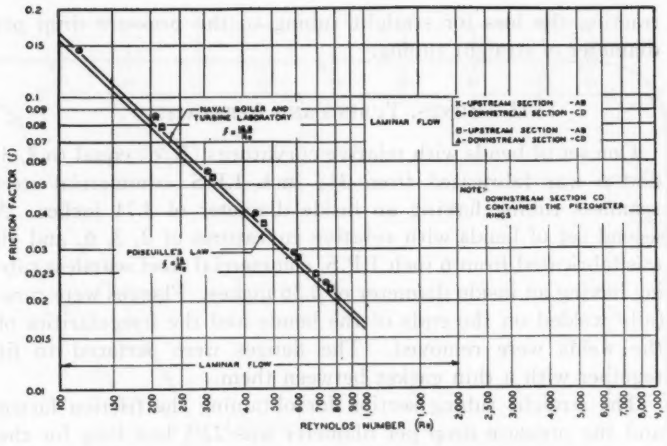

FIGURE 3 - FRICTION FACTOR (f) VS. REYNOLDS NUMBER (R_e) FOR $3\frac{1}{2}$" I.P.S. STRAIGHT TUBING WITH AND WITHOUT PIEZOMETER RINGS

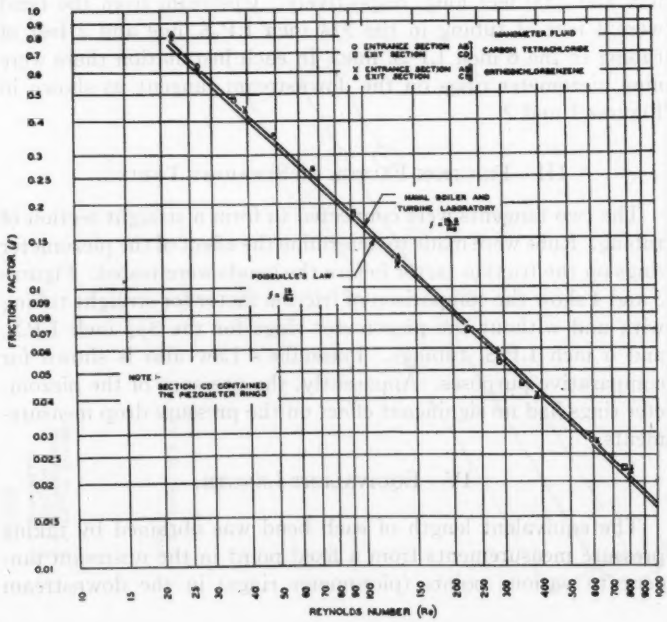

FIGURE 4 - FRICTION FACTOR (f) VS. REYNOLDS NUMBER (R_e) FOR STRAIGHT 6" I.P.S. TUBING WITH AND WITHOUT PIEZOMETER RINGS

tangents. The equivalent length measured in this manner will vary until a point downstream is reached where the disturbance introduced into the fluid by the bend has subsided. The equivalent length measured at or beyond this point is constant and is designated by E_u.

Presented in Table 1 are the E_u values of equivalent length for all the bends of both tubing sizes for the Reynolds numbers run. The corresponding plot is shown in Figure 5. It is apparent that

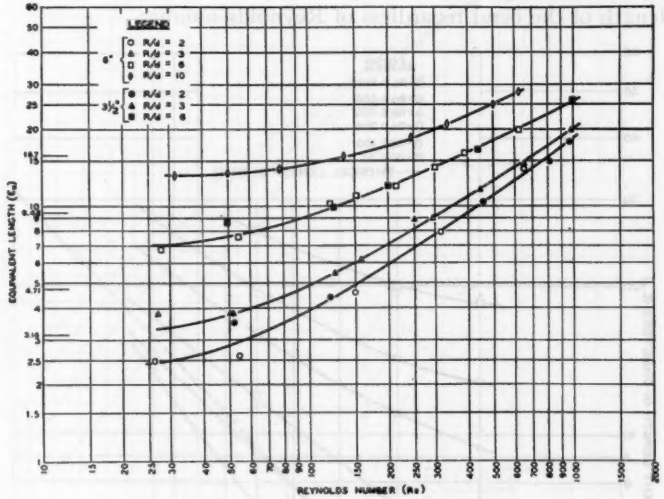

FIGURE 5 - EQUIVALENT LENGTH (E_u) VS. REYNOLDS NUMBER (R_e) FOR 3½" AND 6" I.P.S. BENDS

bends R/d = 2, 3, and 6 give substantially the same curves for both tubing sizes (R/d = 10 was investigated for 6 inch I.P.S. only). This establishes the fact that, for this range of Reynolds numbers, fixing the Reynolds number and the relative radius determines the equivalent length of the bend. Below a Reynolds number of 60 the results were less consistent but the deviations in diameters were small.

In the family of curves representing bends, it appears that for some combinations of Reynolds numbers and relative curvatures, the equivalent length of a bend falls below its physical length.

Indicated to the left is the physical length of each bend in diameters. The Reynolds number at which the equivalent length equals the physical length is called Re_a. This Reynolds number (Re_a) will depend on the shape and position of the curve. For the bends R/d = 2, 3, 6, and 10, respectively, Re_a = 57, 90, 107 and 145. Apparently the higher the relative curvature, the higher is Re_a. However, for some value of R/d there must begin a reversal of this trend since for straight pipe (R/d = ∞), the equivalent length could not possibly be less than the physical length of the bend regardless of Reynolds number.

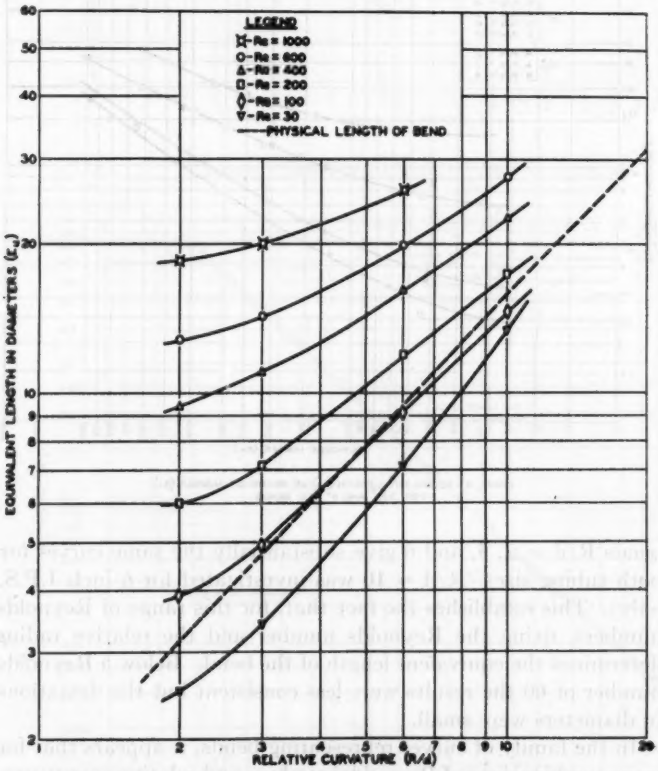

FIGURE 6 - EQUIVALENT LENGTH IN DIAMETERS (E_d) VS. RELATIVE CURVATURE (R/d)

Figure 6 is a cross-plot of Figure 5 showing a family of curves of equivalent length versus relative curvature for constant Reynolds numbers. The dashed line represents the physical length of bends as a function of relative curvature.

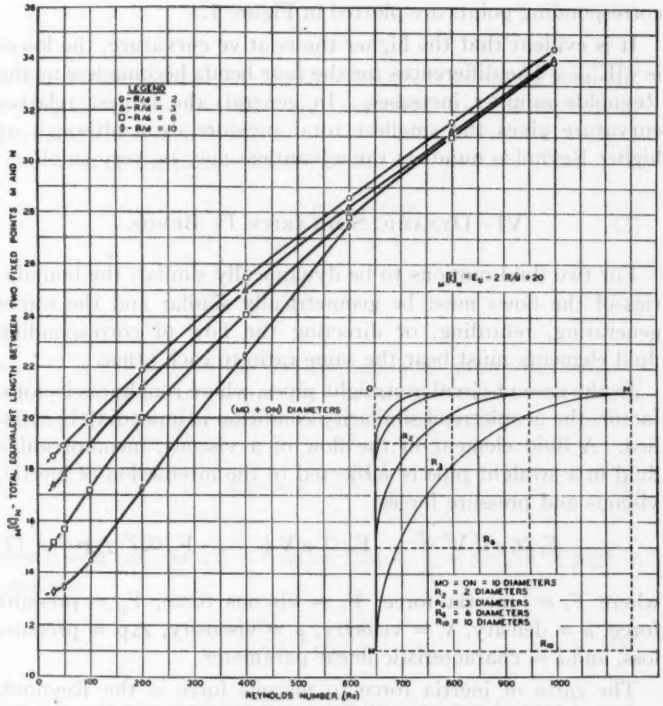
LEGEND
O - R/d = 2
Δ - R/d = 3
□ - R/d = 6
◊ - R/d = 10
(MO + ON) DIAMETERS
MO = ON = 10 DIAMETERS
R2 = 2 DIAMETERS
R3 = 3 DIAMETERS
R6 = 6 DIAMETERS
R10 = 10 DIAMETERS
REYNOLDS NUMBER (Re)

FIGURE 7 - COMPARISON OF TOTAL EQUIVALENT LENGTH OF BEND SYSTEMS FOR OPTIMUM RELATIVE CURVATURE

V—Optimum Relative Curvature.

A right angle bend with constant circular cross-section has the optimum relative curvature when for given flow conditions, the pressure drop in conducting fluid from a point M to a point N is a minimum. Since four bends are under consideration, there are four possible paths for the fluid to take from M to N as

shown in Figure 7. The total equivalent length from M to N is

$$_{M}[E]_{N} = E_u - 2\,R/d + 20 \qquad (1)$$

The calculations from equation (1) are shown in Table 3. The corresponding points are plotted in Figure 7.

It is evident that the higher the relative curvature, the lower is $_{M}[E]_{N}$. The differences for the four bends become less as the Reynolds number increases. In general, the largest relative curvature gives the smallest total pressure drop although at higher Reynolds numbers the advantage may be very small.

VI—Dynamic Similarity In Bends.

For two fluid motions to be dynamically similar, the boundaries of the flows must be geometrically similar and the forces generating, retarding, or directing the flow of corresponding fluid elements must bear the same ratio to each other.

In the case of circular straight pipes, where roughness is not a factor, the geometrical similarity condition is immediately satisfied. A fluid element in the flow of a viscous, incompressible fluid in a straight pipe is subjected to the interaction of inertia, viscous and pressure forces,

$$F_i \propto \rho\, l^2\, V^2 \qquad F_v \propto \mu\, V\, l \qquad F_p \propto l^2\, \Delta p \qquad (2)$$

where F_i = inertia force, F_v = viscous force, F_p = pressure force, ρ = density, V = velocity, μ = viscosity, Δp = pressure loss, and l = characteristic linear parameter.

The ratio of inertia force to viscous force is the Reynolds number which determines the state of fluid flow.

$$\frac{F_i}{F_v} = \frac{\rho\, l^2\, V^2}{\mu\, V\, l} = \frac{\rho\, V\, l}{\mu} \qquad (3)$$

The most pertinent linear parameter is the inside diameter of the pipe, d. Then

$$Re = \frac{\rho\, V\, d}{\mu} \qquad (3a)$$

The ratio of the pressure force to the inertia force gives

$$\frac{F_p}{F_i} = \frac{l^2 \Delta p}{\rho l^2 V^2} = \frac{\Delta p}{\rho V^2} \tag{4}$$

$$P = \frac{\Delta p}{\rho V^2} \tag{4a}$$

where P is the symbol for the dimensionless group formed. (P corresponds to the overall drag coefficient for the motion of bodies immersed in a fluid.)

The ratio of the pressure force to the viscous force gives

$$\frac{F_p}{F_v} = \frac{l^2 \Delta p}{\mu V l} = \frac{l \Delta p}{\mu V} \tag{5}$$

$$N = \frac{l \Delta p}{\mu V} \tag{5a}$$

where N is the symbol for the dimensionless group formed.

The Fanning-D'Arcy equation represents the resistance law of fluid flow in a straight pipe of circular cross-section.

$$\Delta p = k f \rho V^2 \frac{L}{d} \tag{6}$$

where k = a constant, f = friction factor, $\frac{L}{d}$ = length of pipe in diameters.

$$\frac{\Delta p}{\rho V^2} = k f \frac{L}{d} \tag{6a}$$

Since f is a function of the Reynolds number and letting L = d,

$$\frac{\Delta p}{\rho V^2} = k \phi (Re) \tag{6b}$$

$$P \propto \phi (Re) \tag{6c}$$

This fact, of course, could be derived by dimensional analysis. For a straight pipe of constant circular cross-section either Re or P completely characterizes the state of fluid motion. The

magnitude of the Reynolds number represents the relative intensity of the disturbing forces (F_i) and the stabilizing forces (F_v).

In studying the passage of fluids through bends, investigators have generally attempted to correlate the flow phenomena with Reynolds number and the relative curvature. An examination of the results for turbulent flow reported by Hoffman, Vogel, Balch, Davis, Brightmore, Beij, and Nordell shows very little agreement. Since roughness is a very important factor in turbulent flow, it is probable that difference in roughness, as some of these experimenters have suggested, is responsible for these large discrepancies.

The work presented here is entirely in the laminar region where, in straight pipe at least, roughness definitely is not a factor. It is reasonable, then, to assume that, barring the generation of an extremely vigorous disturbance in the fluid, the roughness of the inside surface of the bend or its tangents is not of appreciable importance. Strong support for such a view is given by the data showing the 3½ inch and 6 inch I.P.S. equivalent length curves to be coincident for the same relative curvatures.

When a viscous fluid is forced around a 90 degree bend (or any bend for that matter) a new influence begins to operate on each fluid element. The fluid is now subjected to a centrifugal force (F_c) which is responsible for its curvilinear motion.

If m is the mass of fluid or fluid element,

$$F_c \propto \frac{m V^2}{R} \tag{7}$$

or

$$F_c \propto \frac{\rho l^3 V^2}{R} \tag{7a}$$

Dividing the inertia force by the centrifugal force,

$$\frac{F_i}{F_c} = \frac{\rho l^2 V^2}{\frac{\rho l^3 V^2}{R}} \tag{7b}$$

$$\frac{F_i}{F_c} = \frac{R}{l} \tag{7c}$$

Since for 90 degree pipe bends of constant circular cross-section the simplest linear parameter is the inside diameter, d,

$$\frac{F_i}{F_c} = \frac{R}{d} \tag{7d}$$

The relative curvature of a bend is the ratio of the inertia force to the centrifugal force which, for a given bend, is obviously a constant. Thus, it is impossible to alter the ratio of inertia force to centrifugal force for a given bend.

Dividing the centrifugal force by the viscous force,

$$\frac{F_c}{F_v} = \frac{\frac{\rho l^3 V^2}{R}}{\mu V l} \tag{8}$$

$$\frac{F_c}{F_v} = \frac{\rho l V}{\mu} \cdot \frac{l}{R} \tag{8a}$$

$$\frac{F_c}{F_v} = Re \Big/ \frac{R}{d} \tag{8b}$$

The ratio of the centrifugal to the viscous force is the Reynolds number divided by the relative curvature of the bend.

Dividing the pressure force by the centrifugal force,

$$\frac{F_p}{F_c} = \frac{l^2 \Delta p}{\frac{\rho l^3 V^2}{R}} \tag{9}$$

$$\frac{F_p}{F_c} = \frac{\Delta p}{\rho V^2} \cdot \frac{R}{l} \tag{9a}$$

$$\frac{F_p}{F_c} = P \cdot \frac{R}{d} \tag{9b}$$

Since for dynamic similarity, the force ratios in equations (7d), (8b), and (9b) which govern the flow must be equal, it is obvious that for two bends flows, (1) and (2)

$$\frac{R_1}{d_1} = \frac{R_2}{d_2} \tag{10}$$

$$Re_1 = Re_2 \tag{10a}$$

$$P_1 = P_2 \tag{10b}$$

This means that, excepting forces introduced by differences in roughness, it is necessary and sufficient for dynamic similarity that only two of the three conditions (10), (10a), and (10b) be satisfied.

In the event the flow is turbulent (as is the case in most of the aforementioned investigations) and forces arise from roughness differences all three conditions (10), (10a), and (10b) must be satisfied to insure dynamic similarity. It is imperative that condition (10b) be met since roughness effects, while introducing a new variable, manifest themselves only in the dimensionless group $\frac{\Delta p}{\rho V^2}$. The major problem confronting experimenters in turbulent flow through bends is the difficulty in satisfying condition (10b).

VII—Single Curve for 90 Degree Bends in Laminar Flow.

A new term is now introduced—dynamic equivalence. Two bends are dynamically equivalent if

$$P_1 = P_2 \tag{11}$$

regardless of the relative curvature or Reynolds number. Or

$$\frac{\Delta p_1}{\rho_1 V_1} = \frac{\Delta p_2}{\rho_2 V_2} \tag{11a}$$

From the Fanning-D'Arcy equation

$$\frac{\Delta p}{\rho V^2} = k f \frac{L}{d} \tag{12}$$

In the case of a valve, fitting or bend, $\frac{L}{d}$ is the equivalent length E_u.

$$\frac{\Delta p}{\rho V^2} = k f E_u \tag{12a}$$

$$P = k f E_u \tag{12b}$$

Figure 8 shows a plot of $f E_u$ versus Re where f is the friction factor of the straight tubing corresponding to the Reynolds

number. Since the curves in Figure 8 are not straight lines with a slope of -1, the bends do not act, for this range of Reynolds numbers, in a manner corresponding to Stokes' Law or Poiseuille's Law, i.e., P is not proportional to 1/Re and deformation resistance alone does not govern the flow. It is apparent from the slope of the curves, however, that the larger the relative curvature, the higher will be the Reynolds number at which deforma-

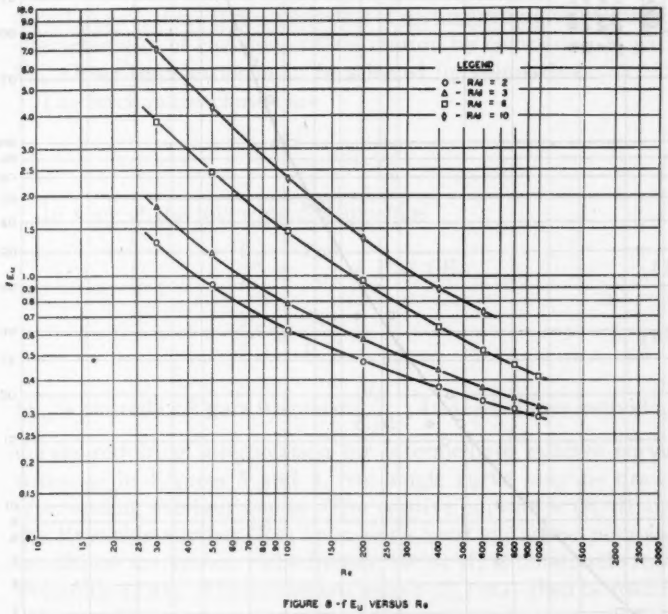

FIGURE 8 - f E_u VERSUS Re

tion resistance fails to dominate the flow. This effect is not unexpected since with increasing relative curvature the bend approaches a straight pipe where resistance is of the deformation type until the lower critical Reynolds number is reached.

In a straight pipe the pressure force, F_p, is related very simply to the viscous force, F_v, since the shearing stress, τ, is proportional to the pressure gradient.

$$\tau = -\frac{dp}{dx} \cdot \frac{r}{2} \tag{13}$$

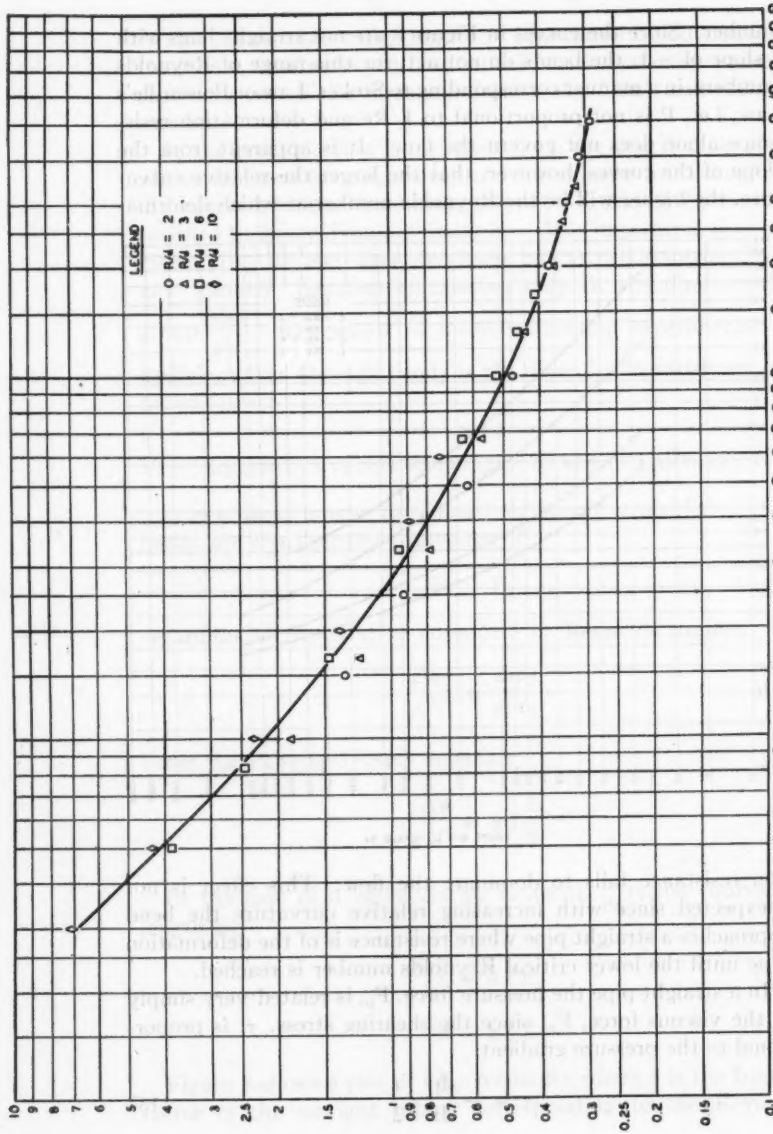

FIGURE 9 - $f\,E_u$ VERSUS $\frac{R_d}{R/d}$

where τ = shearing stress, $\frac{dp}{dx}$ = pressure gradient, r = radial distance from the center of the pipe.

In the case of curvilinear fluid motion, the relation is just as definite but much more complex. A change in the magnitude of the viscous force will require a corresponding although not a proportional change in the pressure force.

Since the inertia force and centrifugal force are both proportional to ρV^2, there is no possibility of distinguishing the dynamic characteristics of curvilinear fluid motion by comparing F_i with F_c. Other force ratios must be selected for comparison.

The force ratios chosen are

$$\frac{F_p}{F_i} \text{ and } \frac{F_c}{F_v}$$

In view of the following relationships

$$\frac{F_p}{F_i} = \frac{\Delta p}{\rho V^2} = P \propto f E_u \tag{14}$$

$$\frac{F_c}{F_v} = \frac{Re}{R/d} \tag{15}$$

$f E_u$ is plotted in Figure 9 against $\frac{Re}{R/d}$. Instead of four individual curves requiring interpolation for intermediate relative curvatures, as in Figures 5 and 8, one single curve may be drawn representing the four bends. The relative curvature (R/d) and the Reynolds number (Re) for a given bend and given pumping conditions are known. The friction factor (f) is obtainable from Poiseuille's Law. The equivalent length (E_u) may then be readily calculated.

TABLE 1

R/d = 2

	3½″ I.P.S.					6″ I.P.S.				
Re	52.0	120	452	819	960	26.2	54.6	149	315	647
E_u	3.6	4.4	10.4	14.9	18.0	2.5	2.6	4.6	7.9	14.0

R/d = 3

	3½″ I.P.S.					6″ I.P.S.				
Re	50.8	122	250	445	980	27.0	53.1	158	294	653
E_u	3.7	5.5	9.0	11.7	20.0	3.7	3.7	6.2	9.0	14.8

R/d = 6

	3½″ I.P.S.					6″ I.P.S.						
Re	49.7	120	199	436	989	27.9	54.6	121	152	214	299	389	620
E_u	8.7	10.0	12.1	16.7	25.8	6.8	7.6	10.1	11.0	12.0	14.1	16.0	20.0

R/d = 10

	6″ I.P.S.							
Re	31.4	50.0	78.1	136	245	338	501	621
E_u	13.6	13.7	14.1	15.8	18.5	21.0	25.2	28.0

TABLE 2

R/d = 2

Re	30	100	200	400	600	1000
E_u	2.5	3.9	6.0	9.4	12.7	18.5

R/d = 3

Re	30	100	200	400	600	1000
E_u	3.4	4.9	7.1	11.0	14.1	20.0

R/d = 6

Re	30	100	200	400	600	1000
E_u	7.1	9.1	11.9	16.0	19.8	25.9

R/d = 10

Re	30	100	200	400	600
E_u	13.3	14.5	17.3	22.6	27.5

TABLE 3

$$_{M}[E]_{N} = E_u - 2\,R/d + 20$$

R/d = 2

Re	30	50	100	200	400	600	800	1000
E_u	2.5	2.9	3.9	5.9	9.5	12.6	15.6	18.4
$_{M}[E]_{N}$	18.5	18.9	19.9	21.9	25.5	28.6	31.6	34.4

R/d = 3

Re	30	50	100	200	400	600	800	1000
E_u	3.4	3.8	4.9	7.2	11.0	14.2	17.2	20.0
$_{M}[E]_{N}$	17.4	17.8	18.9	21.2	25.0	28.2	31.2	34.0

R/d = 6

Re	30	50	100	200	400	600	800	1000
E_u	7.2	7.7	9.2	12.0	16.1	19.7	23.0	25.9
$[E]_{N}$	15.2	15.7	17.2	20.0	24.1	27.7	31.0	33.9

R/d = 10

Re	30	50	100	200	400	600
E_u	13.3	13.5	14.5	17.3	22.5	27.5
$_{M}[E]_{N}$	13.3	13.5	14.5	17.3	22.5	27.5

TABLE 4

R/d = 2

Re	30	50	100	200	400	600	800	1000
E_u	2.5	2.9	3.9	5.9	9.5	12.6	15.6	18.4
f	0.533	0.320	0.160	0.0800	0.0400	0.0267	0.0200	0.0160
f E_u	1.33	0.928	0.624	0.472	0.380	0.336	0.312	0.294

R/d = 3

Re	30	50	100	200	400	600	800	1000
E_u	3.4	3.8	4.9	7.2	11.0	14.2	17.2	20.0
f	0.533	0.320	0.160	0.0800	0.0400	0.0267	0.0200	0.0160
f E_u	1.81	1.22	0.784	0.576	0.440	0.379	0.344	0.320

R/d = 6

Re	30	50	100	200	400	600	800	1000
E_u	7.2	7.7	9.2	12.0	16.1	19.7	23.0	25.9
f	0.533	0.320	0.160	0.0800	0.0400	0.0267	0.0200	0.0160
f E_u	3.84	2.46	1.47	0.960	0.644	0.526	0.460	0.414

R/d = 10

Re	30	50	100	200	400	600
E_u	13.3	13.5	14.5	17.3	22.5	27.5
f	0.533	0.320	0.160	0.0800	0.0400	0.0267
f E_u	7.09	4.32	2.32	1.38	0.900	0.734

TABLE 5

R/d = 2

Re	30	50	100	200	400	600	800	1000
f	0.530	0.320	0.160	0.0800	0.0400	0.0267	0.0200	0.0160
E_u	2.5	2.9	3.9	5.9	9.5	12.6	15.6	18.4
$f E_u \left(\propto \frac{F_p}{F_i}\right)$	1.33	0.928	0.624	0.472	0.380	0.336	0.312	0.294
$Re \Big/ \frac{R}{d} \left(= \frac{F}{F_v}\right)$	15.0	25.0	50.0	100	200	300	400	500

R/d = 3

Re	30	50	100	200	400	600	800	1000
f	0.530	0.320	0.160	0.0800	0.0400	0.0267	0.0200	0.0160
E_u	3.4	3.8	4.9	7.2	11.0	14.2	17.2	20.0
$f E_u \left(\propto \frac{F_p}{F_i}\right)$	1.81	1.22	0.784	0.576	0.440	0.379	0.344	0.320
$Re \Big/ \frac{R}{d} \left(= \frac{F_c}{F_v}\right)$	10.0	16.7	33.3	66.7	133	200	267	333

R/d = 6

Re	30	50	100	200	400	600	800	1000
f	0.530	0.320	0.160	0.0800	0.0400	0.0267	0.0200	0.0160
E_u	7.2	7.7	9.2	12.0	16.1	19.7	23.0	25.9
$f E_u \left(\propto \frac{F_p}{F_i}\right)$	3.84	2.46	1.47	0.960	0.644	0.526	0.460	0.414
$Re \Big/ \frac{R}{d} \left(= \frac{F_c}{F_v}\right)$	5.00	8.33	16.7	33.3	66.7	100	133	167

R/d = 10

Re	30	50	100	200	400	600
f	0.530	0.320	0.160	0.0800	0.0400	0.0267
E_u	13.3	13.5	14.5	17.3	22.5	27.5
$f E_u \left(\propto \frac{F_p}{F_i}\right)$	7.09	4.32	2.32	1.38	0.900	0.734
$Re \Big/ \frac{R}{d} \left(= \frac{F_c}{F_v}\right)$	3.00	5.00	10.0	20.0	40.0	60.0

Glossary of Symbols.

Re	Reynolds number.
f	Friction factor.
E	Equivalent length.
R	Radius of curvature of bend.
d	Inside diameter of bend or tubing.
V	Velocity.
ρ	Density.
μ	Viscosity.
μ/ρ	Kinematic viscosity.
R/d	Relative curvature (or bend ratio).
$_M[E]_N$	Equivalent length from point M to point N for a bend system.
E_u	Real equivalent length, measured to a downstream point outside the transition region.
l	Characteristic linear parameter.
Δp	Pressure drop.
ϕ	Functional symbol.
m	Mass.
F_i	Inertia force.
F_v	Viscous force.
F_p	Pressure force.
F_c	Centrifugal force.
P	F_p/F_i.
N	F_p/F_v.
L	Length of straight tubing.
k	Constant.
$\frac{dp}{dx}$	Pressure gradient.
τ	Shearing stress.
r	Radial distance from center of pipe.
Re_a	Reynolds number at which E_u equals physical length of bend.

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COMPARISON OF 3½ INCH I.P.S. AND 6 INCH I.P.S. VALVES AND FITTINGS IN LAMINAR FLOW.

By Cyrus Beck.*

Note: The opinions or assertions contained in this article are the private ones of the writer and are not to be construed as official or reflecting the views of the Navy Department or the Naval service at large.

I—Introduction.

The problem of extending the graphical and mathematical solutions for particular valves, bends, and fittings—presented in previous articles by the author—to the corresponding valves and fittings of other sizes is important from the standpoint of simplifying the process of securing complete design data. It has been shown experimentally that, where geometrical similarity exists for two bends at the same Reynolds number in the laminar region, the flows are dynamically similar.

The resistance of a valve or fitting to fluid flow is dependent, as was the case with bends, on the relative geometrical contours and the roughness of the valve or fitting and the connecting straight tubing. The degree of roughness is difficult to evaluate. In laminar flow, however, its effect in comparison with the obstruction offered by the physical contour of the valve or fitting is probably small. Assuming this to be the case, conditions may be derived for the geometrical similarity of corresponding valves and fittings for different sizes.

II—Geometrical Similarity.

Assume two flow systems with valves of different sizes and corresponding tubing as shown in Figure 1. For the two systems

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to be geometrically similar, it is necessary that the ratio of all the corresponding dimensions be equal.

Let y_1 and y_2 be the bores of the valves and z_1 and z_2 any other corresponding linear dimensions. Then

$$\frac{z_1}{y_1} = \frac{z_2}{y_2} \tag{1}$$

If equation (1) is satisfied for all linear dimensions, the valves are geometrically similar but the relation of the connecting tubing to the valve must also be considered since equivalent length is in terms of tubing diameters.

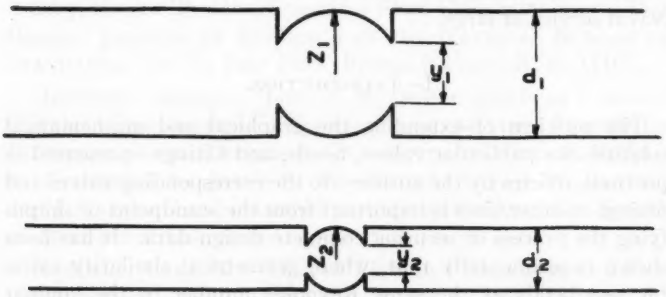

FIGURE 1

Let d_1 and d_2 be the inside diameters of the tubing. Then

$$\frac{y_1}{d_1} = \frac{y_2}{d_2} \tag{2}$$

If equations (1) and (2) are satisfied, the two flow systems are geometrically similar.

III—Geometrical Comparison of Valves and Fittings.

Frequently, investigators make a comparison of equivalent lengths for globe valves. "Globe valves," however, is a generic term covering a variety of dissimilar internal structures which may possibly explain why such widely varying equivalent length values are reported for the "same" valve. The valves compared

may all be "globe" but, unless the flows are dynamically similar, comparison is not properly permissible except as to the relative magnitude of the resistance.

It should be emphasized that the comparison of two valves or fittings of different sizes has meaning only if the general contours of the valves or fittings are the same. For example, the two globe valves tested, 3½ inch I.P.S. and 6 inch I.P.S., have three valve seat ribs which the fluid must encounter in passing through the valve. These valves are comparable. For 3 inch I.P.S. and below, however, these globe valves have no valve seat ribs. Instead they have three disc guides which impede the flow in a different manner. Although they are consecutive sizes in the globe valve series, an exact study of geometrical similarity between globe valves of 3 inch I.P.S. (and below) and 3½ inch I.P.S. (and above) is not possible.

TABLE 1

GLOBE VALVE

Nominal Tubing Size	Tubing I.D.	Bore	Valve Seat	Length	Valve Seat Bore	Length Bore	Bore Tubing I. D.
3½	3.71	4.00	4.375	11.5	1.09	2.88	1.08
6	6.36	6.50	6.875	17.0	1.06	2.62	1.02
8	8.35	8.50	9.000	21.5	1.06	2.53	1.02

It is now necessary to examine the structures of some valves and fittings to determine whether or not they satisfy equations (1) and (2). The valves and fittings to be compared are the 3½ inch, 6 inch, and 8 inch I.P.S. globe valve, angle valve, standard elbow, and tee, all of interest to the Navy Department.

Some pertinent linear dimensions of the globe valve and the conventional connecting tubing are given in Table 1. The three ratios of pertinent dimensions indicate whether or not geometrical similarity exists. Actually, ratios of all the linear dimensions would have to be equal before exact geometrical similarity could be said to obtain. However, the three ratios considered form a good basis for comparison. If any of the corresponding ratios

are not equal, then geometrical similarity does not exist. The closer the ratios are to being equal, the nearer are the valves to the condition of geometrical similarity.

It is apparent that the 6 inch and 8 inch valves satisfy equations (1) and (2) although one pair of ratios are not exactly equal. This means that the equivalent length should be practically the same for both valves. The 3½ inch valve, however, does not satisfy equations (1) and (2) with respect to either the 6 inch or 8 inch valves. This indicates the equivalent length of the 3½ inch valve will not be the same as for the 6 inch and 8 inch valves. The magnitude of the equivalent length difference and the direction cannot be predicted from the data in Table 1.

The foregoing discussion of the globe valve is applicable to the angle valve because the variation in pertinent dimensions with size changes is essentially the same for both.

TABLE 2

STANDARD ELBOW

Nominal Size	Tubing I. D.	Fitting I. D.	Fitting I. D. / Tubing I. D.	R/d
3½	3.71	4.00	1.08	1.125
6	6.36	6.50	1.02	0.962
8	8.35	8.50	1.02	0.882

Table 2 presents some pertinent linear dimensions of the standard elbow and the conventional connecting tubing. In the case of the standard elbow, the 6 inch and 8 inch sizes satisfy equation (2) but not equation (1). This means the equivalent lengths will not be equal although probably fairly close. The 3½ inch elbow does not satisfy equations (1) and (2) with respect to either the 6 inch or 8 inch sizes. Hence, the 3½ inch equivalent length should be expected to differ from the 6 inch and 8 inch valves. Here again how much the equivalent lengths will differ and in what direction cannot be predicted from the data in Table 2.

The discussion of the standard elbow is applicable to the tee because the variation in pertinent dimensions with size changes is approximately the same.

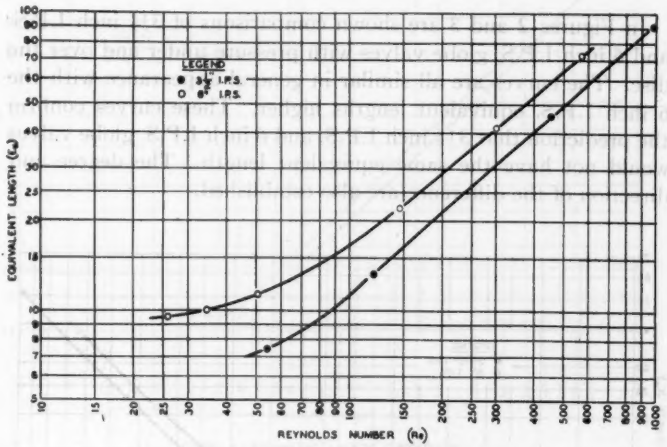

FIGURE 2 - EQUIVALENT LENGTH COMPARISON OF 3½" AND 6" I.P.S. GLOBE VALVE - PRESSURE UNDER DISC

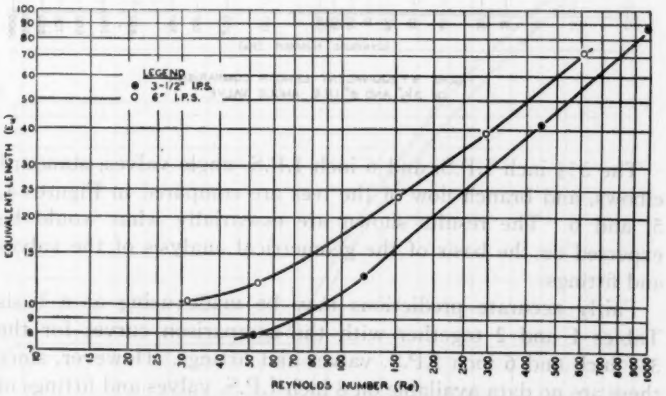

FIGURE 3 - EQUIVALENT LENGTH COMPARISON OF 3½" AND 6" I.P.S. GLOBE VALVE - PRESSURE OVER DISC

IV—Experimental Results.

In Figures 2 and 3 are shown comparisons of 3½ inch I.P.S. and 6 inch I.P.S. globe valves with pressure under and over the disc. The curves are all similar in general appearance with the 6 inch I.P.S. equivalent lengths higher. These curves confirm the prediction that 3½ inch I.P.S. and 6 inch I.P.S. globe valves would not have the same equivalent length. The degree and direction of the difference are also established.

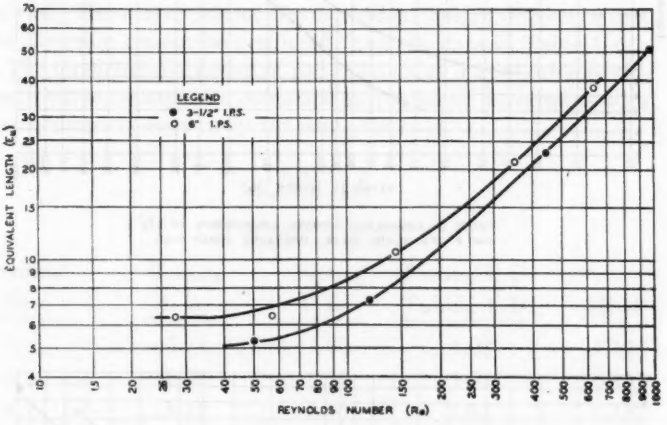

FIGURE 4 - EQUIVALENT LENGTH COMPARISON OF 3½" AND 6" I.P.S. ANGLE VALVE

The 3½ inch I.P.S. and 6 inch I.P.S. angle valves, standard elbows, and branch flow of the tees are compared in Figures 4, 5, and 6. The results shown are essentially what would be expected on the basis of the geometrical analysis of the valves and fittings.

Fairly accurate predictions may be made using as a basis Tables 1 and 2 together with the comparison curves for the 3½ inch and 6 inch I.P.S. valves and fittings. However, since there are no data available on 8 inch I.P.S. valves and fittings of the type being investigated, the validity of such a procedure must await confirmation by further experimental results.

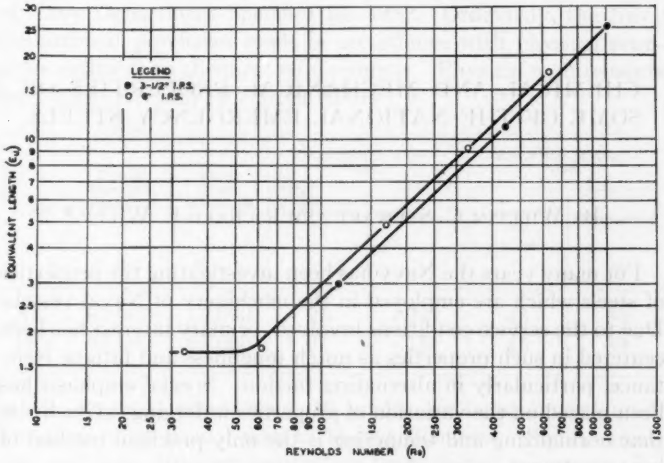

FIGURE 5 - EQUIVALENT LENGTH COMPARISON OF 3½" AND 6" I.P.S. STANDARD ELBOW

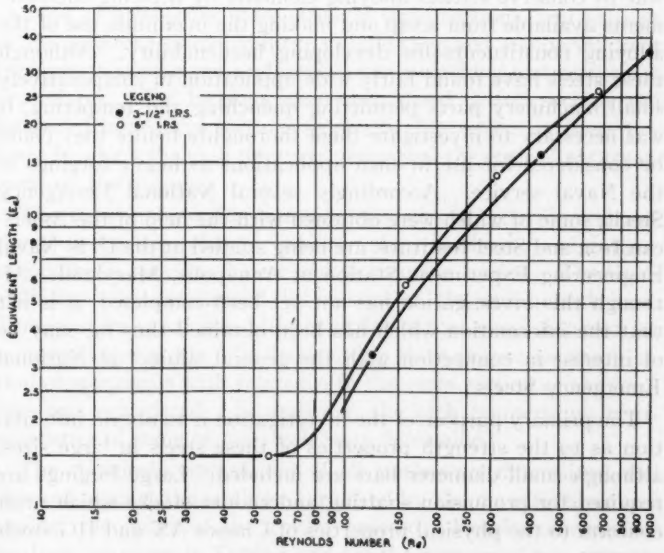

FIGURE 6 - EQUIVALENT LENGTH COMPARISON OF 3½" AND 6" I.P.S. TEE - BRANCH FLOW

CHEMICAL AND MECHANICAL PROPERTIES OF SOME OF THE NATIONAL EMERGENCY STEELS.

By William C. Stewart and Richard E. Wiley.*

For many years the Navy has been investigating the properties of steels which are employed in the machinery of Naval vessels. Due to the service conditions involved, primary interest has been centered in such properties as notch toughness and fatigue resistance, particularly in alternating torsion. Special emphasis has been placed on the variation of properties in forgings of such size that normalizing and tempering is the only practical method of heat treatment.

The National Emergency Steels were developed early in the war to conserve critical alloying elements by utilizing such elements available from scrap and making the maximum use of the alloying constituents for developing hardenability. Although these steels have found fairly wide application in comparatively small machinery parts permitting quenching and tempering, it was necessary to investigate them thoroughly before they could be considered for use in such applications as heavy forgings in the Naval service. Accordingly several National Emergency Steels, some of which were obtained with the help of the American Iron and Steel Institute, are being studied at the U. S. Naval Engineering Experiment Station at Annapolis, Maryland. Although this investigation has not yet been completed, it is felt that the information which has been obtained thus far may be of interest in connection with the general subject of National Emergency Steels.

The primary purpose of the investigation is to obtain information as to the strength properties of these steels in large sizes, although small diameter bars are included. Large forgings are required for propulsion shafting and rudder stocks which must conform to the physical properties of Classes AN and HG steels

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of Navy Department Specification 49S2. Ordinarily, the Navy Department purchases steels in accordance with physical properties rather than chemical requirements. Physical requirements for Classes AN and HG Steels follow:

Class	Tensile Strength Minimum	Proof Stress Minimum	Elongation in 2 ins.—min.		Reduction of Area—min.	
			Longitudinal	Transverse	Longitudinal	Transverse
	P. S. I.	P. S. I.	%	%	%	%
AN	80,000	45,000	25	21	45	30
HG	95,000	65,000	21	18	45	30

In all, 36 steels representing 10 grades are being investigated. The sizes range from 1-inch to 14-inch diameter. Similar steels obtained from different sources are being tested in order to determine the effect of manufacturing variables. In most cases the steels were heat treated by the Engineering Experiment Station. In view of interest in these steels in large sections, and because of the inherent hardenability, normalizing and tempering heat treatments were given preference over quenching and tempering procedures. The forgings were normalized in full section and tempered in as large sections as practicable, except in those few cases in which the manufacturer performed the entire heat treatment on the forging.

The tests include the determination of static tensile and torsional properties, charpy impact value, hardness, and fatigue resistance in rotating flexure and alternating torsion. The hardenability was determined by means of the end-quench test. Temperature-expansion (dilatometric) curves were obtained for representative steels. In determining these properties the location of specimens with reference to the cross section of the forgings was considered. In addition, the high temperature characteristics of some of the steels were determined.

The chemical compositions of all steels were determined and checked against the analyses furnished by the manufacturer and against the range of the grades ordered. Results of chemical analysis are given in Table I and are the average of two or more determinations. It is noted that a considerable number of the

TABLE I

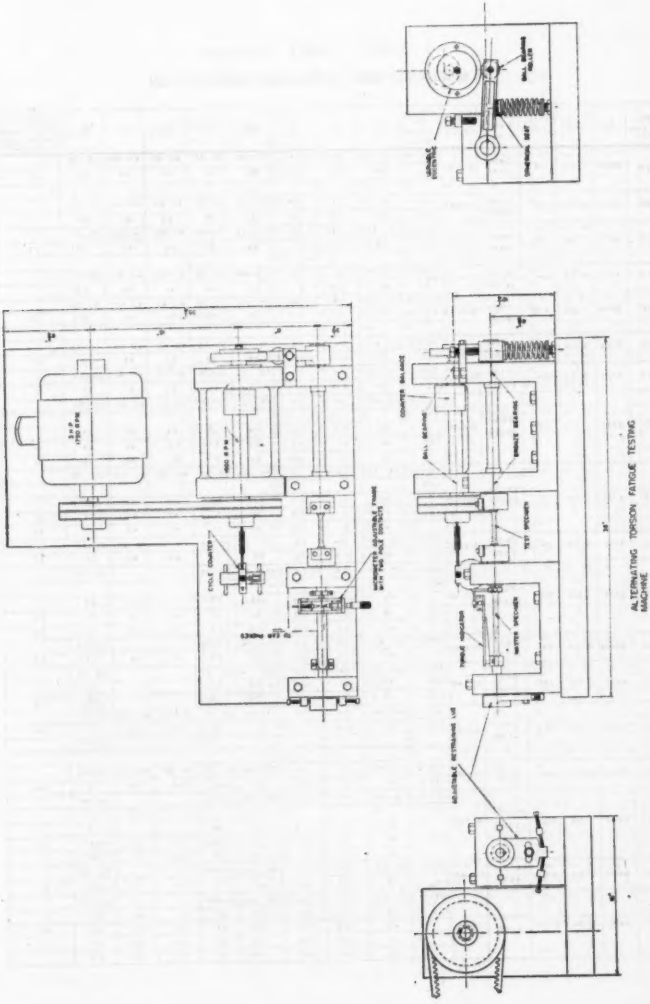
MATERIAL AND CHEMICAL COMPOSITION

MATL LETTER	NE NO.	SIZE	TOTAL LENGTH	NUMBER RANGE (INT)	DATE REC'D	SOURCE OF CHEM RESULTS	C	S	P	Mn	Si	Cr	Ni	Mo	HARDNESS ROCKWELL "C" 1-1/2" FROM END OF JOMINY SPECIMEN
VJ	8749	1" DIA	39 3/8"	5000-5399	3/12/43	SPECS	49-52	.04	.04	75-100	20-35	40-60	40-60	20-30	
						MFR 1									
						2	.48	.027	.018	.76	.26	.53	.48	.23	
						EES 1	.52	.01	.01	.85	.26	.56	.35	.20	34
						2									
WY	8749	6 3/8" DIA	23"	15000-15599	6/18/43	SPECS	.45-.52	.04	.04	75-100	20-35	40-60	40-60	20-30	
						EES	.44	.022	.016	.84	.26	.49	.45	.23	38
XJ	8749	8" DIA	24"	22000-22799	7/24/43	SPECS	.45-.52	.04	.04	75-100	20-35	40-60	40-60	20-30	
						MFR	.48	.019	.023	.96	.25	.49	.51	.23	
						EES	.44	.015	.011	.97	.23	.42	.41	.23	37
WA	8749	1" DIA	39 1/2"	9800-10199	4/20/43	SPECS	.45-.52	.04	.04	75-100	20-35	40-60	40-60	20-30	
						EES 1	.47	.039	.035	.85	.24	.50	.58	.47	50
						2	.40	.049	.02	.90	.22	.50	.60	.46	
						3	.46	.04	.04	.85	.22	.50	.59	.46	
VM	8749 (8949)	8" DIA	25 1/2"	5800-6499	3/30/43	SPECS	.45-.52	.04	.04	75-100	20-35	40-60	40-60	20-30	
						EES	.48	.037	.03	.81	.22	.50	.58	.45	54
WI	8749 (8739)	8" DIA	24"	13400-14199	5/4/43	SPECS	.45-.52	.04	.04	75-100	20-35	40-60	40-60	20-30	
						EES	.40	.012	.021	.97	.24	.43	.47	.27	
XQ	8749	1" DIA	30"	26400-26849	9/25/43	SPECS	.45-.52	.04	.04	75-100	20-35	40-60	40-60	20-30	
						EES 1	.51			.85		.53	.42	.24	32
						2	.50			.76		.50	.45	.20	
						3	.50			.84		.50	.49	.23	
XU	8749	1" DIA	30"	26850-27299	11/4/43	SPECS	.45-.52	.04	.04	75-100	20-35	40-60	40-60	20-30	
						MFR 1									
						2									
						3	.47	.016	.015	.85	.30	.51	.51	.25	
						EES 1	.50	.006	.016	.85	.30	.51	.53	.25	35
						2	.52	.006	.017	.83	.28	.51	.57	.24	
						3	.47	.02	.017	.85	.28	.50	.56	.25	
XD	8949	1" DIA	30"	15200-15599	7/3/43	SPECS	.45-.52	.04	.04	100-130	20-35	40-60	40-60	30-40	
						MFR 1	.45	.018	.023	.110	.23	.54	.50	.36	
						2	.48	.019	.024	.97	.24	.38	.46	.33	
						EES 1	.44	.012	.022	.110	.25	.54	.46	.33	
						2	.44	.012	.022	.90	.25	.34	.43	.32	
XE	8949	8" DIA	24"	16600-17299	7/3/43	SPECS	.45-.52	.04	.04	100-130	20-35	40-60	40-60	30-40	
						EES	.44	.016	.012	.100	.21	.34	.41	.30	37
VZ	8949 (MODIFIED)	1" DIA	40' 8"	9000-9599	4/20/43	SPECS	.45-.52	.04	.04	100-130	20-35	40-60	40-60	30-40	
						EES 1	.48	.022	.03	.84	.20	.58	.56	.34	32
						2	.44	.018	.03	.84	.20	.59	.54	.35	
						3	.38	.023	.02	.86	.20	.58	.54	.34	
VY	8949 (MOD.)	8" DIA	23"	8100-8999	4/30/43	SPECS	.45-.52	.04	.04	100-130	20-35	40-60	40-60	30-40	
						EES	.44	.017	.02	.84	.23	.57	.60	.30	34
VN	8949	1" DIA	32'	6500-6799	4/16/43	SPECS	.45-.52	.04	.04	100-130	20-35	40-60	40-60	30-40	
						EES 1	.45	.007	.022	.100	.22	.46	.35	.32	32
						2									
VD	8949	13 3/4" DIA	24"	1-3699	1/5/43	SPECS	.45-.52	.04	.04	100-130	20-35	40-60	40-60	30-40	
						MFR	.46	.028	.037	.128	.26	.59	.47	.38	
						EES	.49	.032	.022	.110	.34	.54	.51	.37	36
XL	9445 HT46484	8" DIA	36"	24000-25199	7/27/43	SPECS	.43-.48	.04	.04	100-130	40-60	20-40	20-50	08-15	
						EES	.44	.021	.017	.100	.46	.24	.44	.15	34
XI	9445 (9450)	10" DIA	36"	18800-21199	7/23/43	SPECS	.43-.48	.04	.04	100-130	40-60	20-40	20-50	08-15	
						EES	.55	.02	.013	.153	.33	.33	.31	.18	33
XK	9450 HT42129	8" DIA	36"	22800-23999	7/27/43	SPECS	.48-.53	.04	.04	120-150	40-60	20-40	20-50	08-15	
						EES	.53	.032	.020	.126	.45	.22	.21	.13	52

TABLE I (CONTINUED)

MATERIAL AND CHEMICAL COMPOSITION

MATL LETTER	NE NO	SIZE	TOTAL LENGTH	NUMBER RANGE Bar	DATE REC'D	SOURCE OF CHEM RESULTS	C	S	P	Mn	Si	Cr	Ni	Mo	HARDNESS ROCKWELL "C" 1/2" FROM END OF JOMINY SPECIMEN
VK	8630	1" DIA	34"	5400-5799	3/19/43	SPECS	27-33	04	04	70-95	20-35	40-60	40-60	15-25	
						EES 1	28	005	012	86	35	47	36	20	25
						2									
WD	8630	8" DIA	24"	10200-10999	6/14/43	SPECS	27-33	04	04	70-95	20-35	40-60	40-60	15-25	
						EES	30	013	02	83	35	48	40	20	
VI	8630	1" DIA	34"	4800-4999	3/11/43	SPECS	27-33	04	04	70-95	20-35	40-60	40-60	15-25	
						EES 1	28	007	01	89	26	54	42	24	26
						2									
XG	8630	8" DIA	24"	18000-18799	7/17/43	SPECS	27-33	04	04	70-95	20-35	40-60	40-60	15-25	
						EES	30	028	011	75	24	41	36	20	
VX	8739	1" DIA	47 3/4"	7500-8099	4/19/43	SPECS	35-42	04	04	75-100	20-35	40-60	40-60	20-30	
						EES 1	36	01	018	83	30	50	42	22	29
						2									
WE	8739	8" DIA	24"	11000-11799	4/20/43	SPECS	35-42	04	04	75-100	20-35	40-60	40-60	20-30	
						EES	39	011	017	75	28	53	44	22	27
XF	8739	8" DIA	24"	17300-17999	7/3/43	SPECS	35-42	04	04	75-100	20-35	40-60	40-60	20-30	
						EES	39	019	011	93	31	42	41	25	31
XO	8739	1" DIA	30"	25800-26099	8/10/43	SPECS	35-42	04	04	75-100	20-35	40-60	40-60	20-30	
						MFR 1									
						2	40	013	020	98	27	48	48	25	
						EES 1	40	01	012	89	31	49	46	24	31
						2	41	01	004	89	31	51	46	24	31
XP	8739	1" DIA	30"	26100-26399	8/10/43	SPECS	35-42	04	04	75-100	20-35	40-60	40-60	20-30	
						MFR 1									
						2	47	023	018	117	26	48	53	23	
						EES 1	40	01	008	90	31	51	44	22	31
						3	40	011	012	90	31	52	48	25	30
WG	8739	8" DIA	24"	11800-12399	5/4/43	SPECS	35-42	04	04	75-100	20-35	40-60	40-60	20-30	
						EES	40	013	02	88	24	46	40	22	27
XM	8339	1" DIA	30"	25200-25499	8/10/43	SPECS	35-42	04	04	130-160	20-35			20-30	
						MFR 1									
						2	42	019	034	154	22			28	
						EES 1	38	021	013	146	31	16	12	30	33
						2	40	018	013	145	24	29	15	25	35
WH	8338	8" DIA	24"	12600-13599	5/6/43	SPECS	35-42	04	04	130-160	20-35			20-30	
						EES	40	021	018	122	22	17	19	23	29
VO	8338	10" X10"	25"	5800-7499	4/17/43	SPECS	35-42	04	04	130-160	20-35			20-30	
						MFR	44	031	027	159	30			29	
						EES	43	031	021	150	29			26	34
VG	8744	1" DIA	35 1/2"	4000-4599	2/1/43	SPECS	40-47	04	04	75-100	20-35	40-60	40-60	20-30	
						EES 1	43	012	02	91	24	46	42	27	29
						2									
						3									
VF	8744	8" DIA	24"	3200-3999	1/21/43	SPECS	40-47	04	04	75-100	20-35	40-60	40-60	20-30	
						EES	46	01	018	81	40	44	49	24	34
XN	8442	1" DIA	30"	25500-25799	8/10/43	SPECS	38-45	04	04	130-160	20-35			30-40	
						MFR 1									
						2	42	016	024	138	26			33	
						EES 1	42	02	011	139	31	16	12	33	30
						2	42	017	011	143	24	26	13	26	
WJ	8442	8" DIA	24"	14200-14999	5/6/43	SPECS	38-45	04	04	130-160	20-35			30-40	
						EES	46	028	022	125	29	15	10	28	31
VE	8442	1 1/2" DIA	15' 10"	3000-3199	1/9/43	SPECS	38-45	04	04	130-160	20-35			30-40	
						EES 1	38	024	022	122	27	01	04	32	
						2	40	023	021	135	25	01	02	32	30
						3	37	020	016	123	22	01	02	35	
						4	37	021	017	126	21	01	04	38	



ALTERNATING TORSION FATIGUE TESTING MACHINE

FIGURE 1

FIGURE 1

TABLE II

MECHANICAL AND ENDURANCE PROPERTIES OF NATIONAL EMERGENCY STEELS

MATERIAL DESIGNATION	NE DESIGNATION	HEAT TREATMENT: NORMALIZING: SIZE OF SECTION	NORMALIZING: TEMP DEG. FAHR.	NORMALIZING: TIME HELD HOURS	QUENCHING: SIZE OF TEST PIECE	QUENCHING: TEMP DEG. FAHR.	QUENCHING: TIME HELD HOURS	QUENCHING: COOLED	TEMPERING: SIZE OF TEST PIECE	TEMPERING: TEMP DEG. FAHR.	TEMPERING: TIME HELD HOURS	LOCATION AND DIRECTION OF TEST SPECIMENS	MECHANICAL PROPERTIES: TENSILE PROPERTIES: TENSILE STRENGTH (LB. PER SQ. INCH)	PROOF STRESS (LB. PER SQ. INCH)	YIELD STRENGTH 1% SET (LB. PER SQ. INCH)	ELONGATION IN 2 INCHES (PERCENT)	REDUCTION OF AREA (PERCENT)	TORSIONAL PROPERTIES: ULTIMATE STRENGTH (LB. PER SQ. INCH)	TORSIONAL PROPERTIES: PROP. LIMIT (LB. PER SQ. INCH)	TORSIONAL PROPERTIES: MODULUS X10⁶ (LB. PER SQ. INCH)	CHARPY IMPACT FT/LB	BRINELL HARDNESS	ENDURANCE PROPERTIES: ENDURANCE LIMITS: ROTATING CANTILEVER (LB. PER SQ. INCH)	ENDURANCE LIMITS: ALTERNATING TORSION (LB. PER SQ. INCH)	ENDURANCE RATIOS: ENDURANCE LIMIT (FLEXURE) / T.S.	ENDURANCE RATIOS: ENDURANCE LIMIT ALT. TOR. / T.S.
VK	8630	1" DIA	1600	1½					1" DIA	1000	1½		102000	76300	76000	29.5	56.3					198				
													102000	74000	75000	30.0	60.1					201				
												AVERAGE	102000	75150	75750	29.2 -	58.1					200				
										1200			94000	66500	66700	30.0	62.8					181				
													94100	66000	66500	29.5	61.8					185				
												AVERAGE	94050	66250	66600	30.0	62.3					183				
										1500			91250	62500	62600	31.0	62.8					171				
													91250	62000	62900	31.5	61.4					170				
												AVERAGE	91250	62250	62750	31.2	62.1					173				
					1" DIA	1550	1½	OIL	1" DIA	1000	1½		129000	107100	107300	20.0	59.6					290				
													127800	105000	106000	22.5	58.5					280				
												AVERAGE	128400	106050	106650	21.2	59.0					285				
										1200			103800	92500	92600	28.5	66.1					223				
													103800	92000	93000	27.0	65.2					227				
												AVERAGE	103800	92250	92800	27.7	65.6					225				
WD	8630	8" DIA. X 24" LONG	1625	8	QUENCHANT 11" LONG					1200	4	LONGITUDINAL: 1" FROM CENTER	58500	55000	55200	31.0	58.6	67200	58000	12.0	29.6	181 TO 192	40000	26000	.45	.29
												2" FROM CENTER	60000	55000	55200	32.0	58.8				32.8					
												3-1/2" FROM CENTER	58000	54300	54800	31.5	60.1				32.1					
												3-1/2" FROM CENTER	58500	53000	52500	31.5	60.6				30.2					
												AVERAGE	58800	54100	54400	31.5	59.6				32.8					
										1300	4	1" FROM CENTER	54200	52700	53000	33.0	61.1	67300	52000	12.0	31.3	170 TO 176	34000	23000	.43	.28
												2" FROM CENTER	54000	52600	53000	31.5	61.6				30.4					
												3-1/2" FROM CENTER	52500	51400	52000	33.0	63.7				33.6					
												3-1/2" FROM CENTER	51500	52100	52500	33.5	63.5				33.6					
												3-1/2" FROM CENTER	53500	52000	52500	33.5	63.9									
												AVERAGE	53200	52150	52600	32.6	62.7				32.2					
VI	8630	1" DIA	1600	1½					1" DIA	1000	1½		96000	75000	76500	29.5	63.3					194				
													98600	74000	75000	30.0	64.2									
												AVERAGE	98300	75000	75750	29.7	63.7									
										1200			68000	66900	67000	32.0	66.9					183				
													68500	67500	69000	31.5	66.4									
												AVERAGE	68400	67200	68000	31.7	66.6									
										1500			64000	59400	59800	33.5	65.0					161				
													63900	59000	60000	34.0	64.0									
												AVERAGE	63950	59200	59750	33.7	64.5									
					1" DIA	1550	1½	OIL	1" DIA	1000	1½		128000	108700	110000	22.5	63.5					285				
													127300	107000	110000	19.5	63.1									
												AVERAGE	127650	107850	110000	21.0	63.3									

VI	8630	1" DIA	1600	1½								1500		AVERAGE	83950	59300	59750	33.7	64.5										
					1" DIA	1550	1½	OIL	1" DIA	1000	1½			128 000	108700	110000	22.5	63.5					283					
															127300	107000	110000	19.6	63.1										
														AVERAGE	127650	107850	110000	21.0	63.3										
										1200					105500	90800	90600	27.0	71.0					211					
															104800	91000	91500	27.0	70.6										
														AVERAGE	105150	90750	91050	27.0	70.8										
XG	8630	8" DIA. X 24" LONG	1600	0					QUADRANT 11" LONG	1200	4	LONGITUDINAL	2" FROM CENTER	60800	54700	58000	31.0	60.0				276	170 TO 178	56000	24000	.42	.28
													3" FROM CENTER	54500	51300	51500	32.5	60.0				331						
													3-1/2" FROM CENTER	54300	50000	50500	33.0	60.7				336						
													AVERAGE	53850	52000	52350	32.2	60.2				314						
									1" SQ. X 4-3/4"	1200	4	TRANSVERSE	1" FROM OUTSIDE	54500	51200	51500	30.5	48.6				18.8						
													1" FROM OUTSIDE	54000			30.5	47.7				18.0						
													AVERAGE	54250	51200	51500	30.5	48.1				18.4						
VX	8739	1" DIA.	1600	1					1" DIA.	1000	1			113500	92000	93000	24.5	36.7					230					
														116700	92000	92500	24.0	58.2					230					
														AVERAGE	115000	92000	92750	24.2	58.9					230					
										1200					105900	82500	84500	27.5	58.3					204					
															105100	80000	85000	26.5	58.8					210					
														AVERAGE	105000	81500	84750	27.0	58.5					207					
															100400	78000	80000	27.5	60.5					205					
										1300					97400	74000	74500	30.0	63.0					194					
														AVERAGE	98900	76000	77250	28.7	61.9					200					
															98400	61000	82000	30.3	57.8					180					
										1500					99000	62000	64500	29.5	56.0					174					
														AVERAGE	98700	61500	63250	28.9	57.9					177					
					1" DIA.	1550	1	OIL	1" DIA	1200	1			128400	117000	118000	23.8	61.1					268					
															128600	117000	119000	22.7	61.6					270					
														AVERAGE	128450	117000	118500	23.2	61.3					269					
										1300					113300	101500	102500	24.8	63.0					234					
															113800	105600	106500	24.5	62.3					248					
														AVERAGE	113550	103800	104500	24.5	63.1					252					
WE	8739	6" DIA. X 24" LONG	1525	0					QUADRANT 11" LONG	1200	4	LONGITUDINAL	3/8" FROM CENTER									236						
													2" FROM CENTER	85000	52000	52500	32.0	59.4				32.8	174 TO 187	56000	25000	.41	.28
													3-1/2" FROM CENTER	89500	52500	53000	31.5	58.5	64600	28000	12.0	32.1						
													3-1/2" FROM CENTER	89500	53400	54000	31.0	58.0				32.1						
													AVERAGE	88000	52650	53200	31.5	57.5				32.6						
										1500	5		1" FROM CENTER	79200	50700	51000	32.0	62.5				33.6	183 TO 189	53000	24000	.41	.30
													2" FROM CENTER	80800	49200	49600	33.5	62.0				34.4						
													3-1/2" FROM CENTER	82500	53000	53500	32.8	62.0	63600	20000	12.0	33.6						
													3-1/2" FROM CENTER	81000	51900	52000	33.5	61.8				31.3						
													3-1/2" FROM CENTER	81700	47000	49000	33.0	62.5										
													AVERAGE	81100	50250	51000	33.0	62.2				33.5						

TABLE II (CONTINUED)

MECHANICAL AND ENDURANCE PROPERTIES OF NATIONAL EMERGENCY STEELS

MATERIAL DESIGNATION	NE DESIGNATION	HEAT TREATMENT: NORMALIZING: SIZE OF SECTION	NORMALIZING: TEMP DEG. FAHR.	NORMALIZING: TIME HELD HOURS	QUENCHING: SIZE OF SPECIMEN	QUENCHING: TEMP DEG. FAHR.	QUENCHING: TIME HELD HOURS	QUENCHING: COOLED	TEMPERING: SIZE OF SPECIMEN	TEMPERING: TEMP DEG. FAHR.	TEMPERING: TIME HELD HOURS	LOCATION AND DIRECTION OF TEST SPECIMENS	MECHANICAL PROPERTIES: TENSILE PROPERTIES: TENSILE STRENGTH (LB. PER SQ. INCH)	PROOF STRESS (LB. PER SQ. INCH)	YIELD STRENGTH .1% SET (LB. PER SQ. INCH)	ELONGATION IN 2 INCHES (PERCENT)	REDUCTION OF AREA (PERCENT)	TORSIONAL PROPERTIES: ULTIMATE STRENGTH (LB. PER SQ. INCH)	TORSIONAL PROPERTIES: PROP. LIMIT (LB. PER SQ. INCH)	TORSIONAL PROPERTIES: MODULUS X10⁶ (LB. PER SQ. INCH)	CHARPY IMPACT FT/LB	BRINELL HARDNESS	ENDURANCE PROPERTIES: ENDURANCE LIMITS: ROTATING CANTILEVER (LB. PER SQ. INCH)	ENDURANCE LIMITS: ALTERNATING TORSION (LB. PER SQ. INCH)	ENDURANCE RATIOS: ENDURANCE LIMIT (CANTILEVER) / T.S.	ENDURANCE RATIOS: ENDURANCE LIMIT (ALT. TOR.) / T.S.
XF	8739	8" DIA X 24" LONG	1600	8	QUADRANT 11" LONG				QUADRANT 11" LONG	1200	5	LONGITUDINAL: 1" FROM CENTER	87000	81000	81500	28.0	58.8	80000	30000	11.0	228	179 TO 197	37000	28000	.42	.28
												LONGITUDINAL: 2" FROM CENTER	87500	83500	84000	28.5	62.8				228					
												LONGITUDINAL: 3-1/2" FROM CENTER	90800	86600	86050	30.0	62.5				236					
												LONGITUDINAL: AVERAGE	88400	84700	83150	29.1	61.7				233					
					1" SQ. X 4-3/4" LONG				1" SQ. X 4-3/4" LONG			TRANSVERSE: 1" FROM OUTSIDE	89300	87300	87700	28.0	55.0									
												TRANSVERSE: 1" FROM OUTSIDE	89500	87200	87800	28.0	54.7									
												TRANSVERSE: AVERAGE	89900	87250	87750	28.5	53.8									
					QUADRANT 11" LONG				QUADRANT 11" LONG	1300	4	LONGITUDINAL: 1" FROM CENTER	89000	83900	83000	28.0	58.6	69700	28400	12.0	30.9	177 TO 192	34000	25000	.36	.26
												LONGITUDINAL: 2" FROM CENTER	88000	83000	83400	30.5	60.8									
												LONGITUDINAL: 3" FROM CENTER	92200	61900	62300	28.5	61.8				32.1					
												LONGITUDINAL: AVERAGE	89700	83950	84400	29.0	60.6				31.3					
					1" SQ. X 4-3/4" LONG				1" SQ. X 4-3/4" LONG			TRANSVERSE: 1" FROM OUTSIDE	90500	86900	87400	28.0	51.9									
												TRANSVERSE: 1" FROM OUTSIDE	92500	86600	87000	23.5	36.0									
												TRANSVERSE: AVERAGE	91500	86750	87200	28.7	43.9									
XO	8739	1" DIA.	1600	1					1" DIA.	1100	1		120000	95400	96000	23.0	54.9									
													127200	103100	103500	21.0	53.0									
												AVERAGE	123000	99250	99750	22.0	53.9									
										1200			112500	91600	92050	25.0	57.8									
													118500	94700	95000	22.0	55.4									
												AVERAGE	115500	93150	93500	23.5	56.6									
										1300			115000	90000	90500	21.5	58.7									
													114500	91700	92200	22.0	58.0									
												AVERAGE	114750	90850	91350	21.7	58.8									
										1500			97000	66500	67500	30.5	61.8									
													97000	63700	65500	31.0	61.5									
												AVERAGE	97000	65100	66500	30.7	61.7									
										1100	1		154000	143700	149000	18.5	52.0									
													154000	140000	145000	19.0	54.4									
												AVERAGE	154000	141800	145000	18.8	53.2									
					1" DIA.	1550	1	OIL	1" DIA.	1200	1		133500	129000	129000	21.5	58.1									
													135000	126000	127000	21.0	57.7									
												AVERAGE	134250	127000	128000	21.2	57.9									
										1300			108500	98000	100000	27.0	62.4									
													112200	104000	106000	28.5	61.7									
												AVERAGE	110350	101500	103000	28.7	62.3									
										1100			121700	98800	99300	22.5	56.4									
													122000	93700	94000	20.5	53.5									

| | | |

TABLE II (CONTINUED)

MECHANICAL AND ENDURANCE PROPERTIES OF NATIONAL EMERGENCY STEELS

MATERIAL DESIGNATION	NE DESIGNATION	HEAT TREATMENT: NORMALIZING: SIZE OF SECTION	NORMALIZING: TEMP DEG FAHR	NORMALIZING: TIME HELD HOURS	QUENCHING: SIZE OF SPECIMEN	QUENCHING: TEMP DEG FAHR	QUENCHING: TIME HELD HOURS	QUENCHING: QUENCHED	TEMPERING: SIZE OF SPECIMEN	TEMPERING: TEMP DEG FAHR	TEMPERING: TIME HELD HOURS	LOCATION AND DIRECTION OF TEST SPECIMENS	MECHANICAL PROPERTIES: TENSILE PROPERTIES: TENSILE STRENGTH	PROOF STRESS	YIELD STRENGTH 1% SET	ELONGATION IN 2 INCHES	REDUCTION OF AREA	TORSIONAL PROPERTIES: ULTIMATE STRENGTH	PROP. LIMIT	MODULUS X10⁶	CHARPY IMPACT FT/LB	BRINELL HARDNESS	ENDURANCE PROPERTIES: ENDURANCE LIMITS: ROTATING CANTILEVER	ENDURANCE LIMITS: ALTERNATING TORSION	ENDURANCE RATIOS: ENDURANCE LIMIT (FLEXURE) T.S	ENDURANCE RATIOS: ENDURANCE LIMIT ALT TOR T.S
													LB PER SQ INCH			PERCENT		LB PER SQ INCH			FT/LB		LB PER SQ INCH			
XM	8339	1" DIA	1600	1					1" DIA.	1100	1		115700	93700	84000	22.0	56.8									
													110000	90000	90500	24.0	56.2									
												AVERAGE	112850	91850	92250	23.0	57.5									
										1200			102500	82500	83000	25.5	54.5									
													108000	88400	89000	23.0	58.0									
												AVERAGE	109750	85450	86000	24.2	56.2									
										1300			104500	81000	81500	23.0	58.8									
													96000	76900	77500	25.5	54.8									
												AVERAGE	100250	78950	79500	24.2	62.3									
										1500			98700	68100	68500	25.5	62.3									
													100200	67500	68000	32.0	62.9									
												AVERAGE	99450	67800	68250	30.7	62.6									
					1" DIA.	1550	1	OIL	1" DIA.	1100	1		143800	133700	134400	19.0	57.0									
													134800	142500	143000	17.0	54.9									
												AVERAGE	140750	138100	138700	18.0	55.9									
										1200			117000	106200	106800	24.0	63.2									
													118500	106400	109000	24.0	63.5									
												AVERAGE	117750	107300	107900	24.0	63.8									
										1300			102500	93900	96500	29.5	66.7									
													125000	115600	116000	20.0	61.7									
												AVERAGE	113750	105750	106250	24.7	64.2									
WH	8339	6" DIA. X 24" LONG	1625	8	QUADRANT 11" LONG					1000	4	LONGITUDINAL	1" FROM CENTER	105000	70000	72500	24.5	53.0	76000	25000	11.9	26.0	223 TO 241	51000	31000	.47	.29
													2" FROM CENTER	107500	75500	75600	25.0	53.0				29.6					
													3" FROM CENTER	111000	78500	80000	23.0	51.4				23.3					
													3" FROM CENTER	110700	79500	80000	25.5	50.8									
												AVERAGE	108850	75850	77000	25.2	52.0				26.6					
										1200			1" FROM CENTER	91000	63000	64000	30.0	61.8	71100	21000	11.8	33.2	194 TO 204	43000	26500	.46	.29
													2" FROM CENTER	92200	66000	66200	28.0	61.6				32.8					
													3" FROM CENTER	93500	67900	68100	31.0	60.8				34.4					
													3" FROM CENTER	93700	64300	68700	30.5	61.2				33.1					
													3" FROM CENTER	94500	68300	69000	28.5	60.7									
												AVERAGE	93400	66350	67300	29.8	61.2				33.9					
										1360			1" FROM CENTER	87700	54000	55000	30.5	59.0	69300	18000	12.1	40.0	179 TO 191	42000	24000	.47	.27
													2" FROM CENTER	89000	59700	60500	34.5	60.3				36.0					
													2-1/2" FROM CENTER	94300	61300	62200	29.0	58.8				38.2					

WH	8339	8" DIA. X 24" LONG	1625	8				4	LONGITUDINAL	3" FROM CENTER	96300	66900	69000	28.5	60.7									
										AVERAGE	93400	68350	67300	29.8	61.2									
										1" FROM CENTER	87700	54000	55000	30.5	58.0									
										2" FROM CENTER	89000	59700	60300	31.5	60.3									
										3-1/2" FROM CENTER	94300	61300	62200	29.0	58.8	69300	18000	12.1	352	179 TO 191	42000	24000	.47	.27
										5-1/2" FROM CENTER	89000	63500	63000	32.0	63.8				327					
										AVERAGE	89950	59600	60400	30.7	60.5				336					
										1" FROM CENTER	92500	57000	57500	29.5	52.5				27.6					
										2" FROM CENTER	94500	58100	60000	30.0	53.5				27.6					
										3-1/2" FROM CENTER	99000	63100	63500	27.5	53.6	70800	19000	12.0	29.4	187 TO 208	41000	25000	.43	.26
										5-1/2" FROM CENTER	98800	59000	59700	27.5	54.0				23.3					
										AVERAGE	98490	59550	60100	28.6	53.3				26.1					
VO	8339	10" SQ. X 24" LONG	NORMALIZED AND TEMPERED BY MANUFACTURER						LONGITUDINAL	POSITION 2	120500	54000	75000	8.0 *	43					235 TO 260	45000		.38	
										POSITION 2	119500	55000	75000	8.0 *	43									
										POSITION 9	119800	66700	76000	13.7	15.6									
										POSITION 9	119200	66700	77500	7.8	9.6									
										POSITION 13	120500	56500	76000	20.0	41.9				12.8					
										POSITION 16	120000	58000	76000	17.5	35.3									
										POSITION 16	120500	56000	76000	18.0	36.3									
										AVERAGE	119900	58950	75950	12.6	20.9				12.8					
									TRANSVERSE	1" FROM SURFACE	120000	96500	79000	12.0	11.8				9.3					
										4" FROM SURFACE	111000	91200	72500	5.0	5.4				8.7					
										2" FROM SURFACE	114000	91000	76500	4.5	3.9				4.6					
										4" FROM SURFACE	112000	53000	76000	4.0	4.3				3.8					
										45° TO SURFACE	121500	96500	76750	9.5	9.6				8.2					
										45° TO SURFACE	122200	93700	76200	9.5	11.5				8.9					
										AVERAGE	116800	93750	76500	7.1	7.8				7.3					
VG	8744	1" DIA.	1600	1½							140700	98400	103100	18.0	48.1					292				
											141000	97800	101000	18.5	46.0									
										AVERAGE	140850	97850	102050	18.7	48.5									
											128700	96200	98400	18.5	51.7					262				
											128300	96000	98000	20.5	51.5									
										AVERAGE	128450	96100	98200	20.0	51.4									
											122500	70000	70500	22.0	44.5					244				
											122500	70000	71000	21.5	46.0									
										AVERAGE	122500	70000	70750	22.2	45.1									
											166300	161200	172500	10.5	41.0					296				
											166500	169000	174000	18.0	44.8									
										AVERAGE	166400	165100	173250	19.2	42.9									
											140500	128000	129500	20.5	54.9					296				
											142500	130000	131000	20.5	56.9									
										AVERAGE	141500	129400	130250	20.5	55.7									

* BROKE OUTSIDE PUNCH MARKS

TABLE II (CONTINUED)

MECHANICAL AND ENDURANCE PROPERTIES OF NATIONAL EMERGENCY STEELS

MATERIAL DESIGNATION	NE DESIGNATION	HEAT TREATMENT: NORMALIZING: SIZE OF SECTION	NORMALIZING: TEMP DEG. FAHR.	NORMALIZING: TIME HELD HOURS	QUENCHING: SIZE OF TEST SPECIMEN	QUENCHING: TEMP DEG. FAHR.	QUENCHING: TIME HELD HOURS	QUENCHING: COOLED	TEMPERING: SIZE OF TEST SPECIMEN	TEMPERING: TEMP DEG. FAHR.	TEMPERING: TIME HELD HOURS	LOCATION AND DIRECTION OF TEST SPECIMENS	MECHANICAL PROPERTIES: TENSILE PROPERTIES: TENSILE STRENGTH (LB PER SQ. INCH)	PROOF STRESS (LB PER SQ. INCH)	YIELD STRENGTH .1% SET (LB PER SQ. INCH)	ELONGATION IN 2 INCHES (PERCENT)	REDUCTION OF AREA (PERCENT)	TORSIONAL PROPERTIES: ULTIMATE STRENGTH (LB PER SQ. INCH)	TORSIONAL PROPERTIES: PROP. LIMIT (LB PER SQ. INCH)	TORSIONAL PROPERTIES: MODULUS X10⁶ (LB PER SQ. INCH)	CHARPY IMPACT FT/LB	BRINELL HARDNESS	ENDURANCE PROPERTIES: ENDURANCE LIMITS: ROTATING CANTILEVER (LB PER SQ. INCH)	ENDURANCE LIMITS: ALTERNATING TORSION (LB PER SQ. INCH)	ENDURANCE RATIOS: ENDURANCE LIMIT (ROTATING) T.S.	ENDURANCE RATIOS: ENDURANCE LIMIT ALT TOR. T.S.
VF	8744	8" DIA. X 36 LONG	1400	4					QUADRANT 11" LONG	1000	5	LONGITUDINAL	117700	67500	70500	21.5	46.6				15.4	207 TO 217	46000	33000	.39	.28
													117000	68000	70000	21.0	48.1				15.2					
													122000	73000	77000	21.5	42.0	93100	38050	12.2	19.2					
													117000	70500	77000	21.2	46.9									
													118400	69750	75900	21.2	46.1				17.3					
										1100			110700	66000	72000	20.6	42.8				16.7					
													119900	68900	72000	23.0	44.6				18.9					
													119000	71000	73500	21.6	43.4	91600	38000	12.0		212 TO 223	47000	32000	.39	.27
													120300	74000	76500	21.3	44.5									
													119700	69900	73500	21.6	44.3				18.3					
										1275			105000	61000	62500	24.5	48.5				20.0					
													102800	63800	65000	26.2	53.6				21.9					
													107300	69000	70000	25.5	54.3	79100	38500	12.0	24.7	205 TO 207	43000	31500	.41	.30
													103500	69000	70000	25.0	54.1				25.3					
													105400	65600	66900	25.3	52.4				22.6					
										1500			105500	59500	56500	23.0	42.0				16.4					
													101900	57000	57500	25.0	46.8				16.7					
													100200	57000	58000	27.5	48.1	72400	38000	12.1	19.0	176 TO 179	41000	27000	.40	.27
													99400	53500	55500	28.0	47.5				18.9					
													101750	58750	56900	26.0	46.5				18.0					
XN	8442	1" DIA.	1600	1					1" DIA.	1100	1		116500	91500	92000	23.0	59.5									
													115500	92000	92500	23.0	57.5									
												AVERAGE	116000	91750	92250	23.0	58.5									
										1200			113300	91000	91500	22.0	62.0									
													106500	86000	86500	26.5	61.8									
												AVERAGE	110000	88500	89000	24.2	61.9									
										1300			97500	78400	78600	29.0	65.2									
													108500	83600	84000	24.5	61.7									
												AVERAGE	103000	81100	81400	26.7	63.4									
										1500			103000	70400	70700	29.5	58.1									
													101000	71300	71500	29.0	60.0									
												AVERAGE	102000	70850	71100	29.2	59.0									
					1" DIA.	1550	1	OIL	1" DIA.	1100	1		147500	136500	139500	21.0	59.1									
													162500	153000	153800	16.5	57.0									
												AVERAGE	155500	145000	146650	18.7	58.0									
										1200			124500	120000	120500	25.0	63.3									
													126000	120000	121000	24.0	63.7									
												AVERAGE	125250	120000	120750	24.5	63.5									

												1100		163500	155000	153800	16.5	67.0													
													AVERAGE	155500	145900	146680	12.7	58.0													
														124500	120000	120500	25.0	63.5													
						1" DIA.	1250	1	OIL	1" DIA.	1200	1		126000	120000	121000	24.0	63.7													
													AVERAGE	125250	120000	120750	24.5	63.5													
														114500	107500	108500	23.0	66.0													
												1300		106300	99200	99900	30.0	66.7													
													AVERAGE	109900	103350	104000	26.5	66.3													
													1" FROM CENTER	105000	76500	77500	25.0	54.1				223 TO 241								
													1-1/2" FROM CENTER	107400	75000	77400	24.5	51.8													
												1000	3" FROM CENTER	112500	80600	82800	20.5	51.7	60100	20000	11.9		48000	33000	.44	.30
													3-1/2" FROM CENTER	113000	82100	83400	22.0	51.9				21.2								
													AVERAGE	108700	78800	80300	22.9	52.4				24.2								
													1" FROM CENTER	91300	64000	64400	28.5	60.1				32.1								
													2" FROM CENTER	96500	71200	71600	27.5	58.5				28.6								
												1250	3" FROM CENTER	97200	68000	70000	26.0	58.6	73100	20000	11.9	28.9	197 TO 210	43000	28000	.45	.29
													3-1/2" FROM CENTER	96500	66500	69000	28.5	60.3				32.1								
	WJ	8442	8" DIA X 24" LONG	1625	8								AVERAGE	96300	68180	68800	28.6	59.7				31.7								
													1" FROM CENTER	90000	57000	57500	30.0	60.3				32.1								
													2" FROM CENTER	92500	58700	59000	29.0	58.4				34.4								
												1300	3-1/4" FROM CENTER	88000	61800	62500	30.0	61.7	71600	21000	12.0		185 TO 197	41000		.45	
													3-1/4" FROM CENTER	94000	59500	61500	30.0	59.3				27.8								
													3-1/2" FROM CENTER	88500	58700	58200	32.0	62.0				28.6								
													AVERAGE	90600	59150	59900	30.2	60.8				30.6								
													1" FROM CENTER	93000	58200	58600	29.0	54.1				24.4								
													2" FROM CENTER	96500	60000	60500	28.0	50.8				26.1								
												1500	3-1/4" FROM CENTER	97500	59000	60000	27.5	52.7	71200	28000	12.2		195 TO 207	41000	27000	.44	.29
													3-1/2" FROM CENTER	98000	62100	62500	30.0	53.0				23.6								
													AVERAGE	96250	59800	60400	28.8	52.6				24.7								
														108700	71000	79800	28.0	52.8					215							
										1-1/4"	1500	1		109000	73500	76500	26.5	54.8					218							
													AVERAGE	108850	72250	78750	26.2	54.8					216							
														142500	110000	113000	17.0	55.5					316							
	VE	8442	1-1/2" DIA.	1600	1							1000		136000	105000	106000	18.5	54.9					316	75000		.54	
						1-1/2" DIA.	1500	1	OIL	1-1/4"		1	AVERAGE	139250	107500	109500	17.8	55.2					316							
														114700	93000	95000	25.0	66.1				23.6	241							
												1300		118000	93000	94000	22.0	64.7				22.0	243	56000	36000	.50	.31
													AVERAGE	116350	93000	94500	23.5	65.4				22.8	242							
														126500	95500	96600	21.0	53.3					248							
												1000		124700	92000	93500	22.0	52.8					239							
													AVERAGE	125600	93750	94950	21.5	53.0					256							
														113000	83000	83900	24.0	56.8					220							
										1" DIA.	1800	1-1/2		112500	84000	85000	24.5	57.5					222							
													AVERAGE	112750	83950	84450	24.2	57.1					220							

										3-1/2" FROM CENTER	117000	70500	77000	21.2	46.9				12.3	217				
										AVERAGE	118400	69750	75900	21.2	46.1									
										1" FROM CENTER	119700	66000	75000	20.8	45.9				14.7					
										2" FROM CENTER	119900	68900	72000	23.0	44.6				19.9					
								1100		3-1/2" FROM CENTER	116000	71000	73500	21.5	45.4	81800	28800	12.0		212 TO 283	47000	32000	.38	.27
										3-1/2" FROM CENTER	120300	74000	75800	21.3	44.5									
										AVERAGE	119700	69900	73300	21.5	44.3				18.3					
VF	8744	8" DIA. X 84 LONG	1600	4						1" FROM CENTER	105000	64000	62500	24.5	45.5				20.8					
										2" FROM CENTER	106800	63500	63000	24.2	53.6				21.9					
								1275	3	3-1/2" FROM CENTER	107300	66000	70000	23.5	54.3	75100	28500	12.0	24.7	200 TO 287	43000	31500	.41	.30
										3-1/2" FROM CENTER	103500	69000	70000	25.0	56.1				23.3					
										AVERAGE	108400	63900	66900	23.3	52.4				22.6					
										1" FROM CENTER	105500	59500	56500	23.0	43.0				16.4					
										2" FROM CENTER	101500	57000	57500	25.2	46.8				16.7					
								1500		3-1/2" FROM CENTER	100200	57000	58000	27.5	48.1	72400	28000	12.1	19.0	176 TO 179	41000	27000	.40	.27
										3-1/2" FROM CENTER	99400	53500	55500	28.0	47.5				18.8					
										AVERAGE	101750	56750	56900	26.0	48.5				18.0					
											116500	91500	92000	23.0	58.5									
								1100			115500	92000	92500	23.0	57.5									
										AVERAGE	116000	91750	92250	23.0	58.5									
											113300	91000	91500	22.0	62.0									
								1200			106500	86000	86500	26.5	61.8									
							1" DIA.		1	AVERAGE	110900	88500	89000	24.2	61.9									
											97500	78400	78600	29.0	63.2									
								1300			108500	83800	84000	24.5	61.7									
										AVERAGE	103000	81100	81400	26.7	63.4									
											103000	70600	70700	29.5	58.1									
XN	8442	1" DIA.	1500	1				1500			101000	71300	71500	29.0	60.0									
										AVERAGE	102800	70950	71100	29.2	59.0									
											167500	138800	139500	21.0	59.1									
								1100			163500	153000	153800	16.5	57.0									
										AVERAGE	155500	145900	146650	18.7	58.0									
											124500	120000	120500	25.0	63.3									
					1" DIA.	1550	1	OIL	1" DIA.	1200	126000	120000	121000	24.0	63.7									
										AVERAGE	125250	120000	120750	24.5	63.5									
											114500	107500	108500	23.0	66.0									
								1300			108300	99500	99500	30.0	66.7									
											109900	103350	104000	26.5	66.3									
										1" FROM CENTER	106000	76500	77500	25.0	54.1				22.0					
										1-1/2" FROM CENTER	107400	78000	77400	24.0	51.9				24.7					
								1000		3" FROM CENTER	112500	80600	82600	20.5	51.7	80100	30000	11.9	21.9	223 TO 241	48000	33000	.44	.30
										3-1/2" FROM CENTER	113000	83100	83000	22.0	51.9				21.2					
										AVERAGE	109700	79800	80300	22.9	52.4				24.2					
										1" FROM CENTER	91300	64000	64400	28.5	60.1				32.1					
										2" FROM CENTER	98600	71200	71800	27.5	59.8				32.0					
								1200		3" FROM CENTER	97300	68000	70000	28.0	58.6	73100	20000	11.9	28.9	197 TO 210	43000	28000	.45	.28
										3-1/2" FROM CENTER	96800	68500	69000	29.5	60.5				32.1					

														AVERAGE	108700	78600	80300	22.9	52.4				54.2				
														1" FROM CENTER	81200	64000	64400	28.5	62.1				52.1				
														3" FROM CENTER	96500	71500	71800	27.5	58.8				52.8				
														5" FROM CENTER	97200	68000	70000	28.0	58.6	73100	20000	11.9	52.9	197 TO 210	43000	58000	.45	.28
														5-1/8" FROM CENTER	96500	68500	68000	28.5	60.3				52.1				
														AVERAGE	95300	68100	68800	28.6	58.7				51.7				
WJ	8442	8" DIA. X 24" LONG	1685	8						11" LONG	1200	4	LONGITUDINAL	1" FROM CENTER	90000	57000	57300	30.0	60.3				52.1				
														3" FROM CENTER	92500	58700	59000	29.0	59.4				54.4				
														5-1/8" FROM CENTER	89000	61500	62500	30.0	61.7	71800	21000	12.0		188 TO 197	41000		.45	
											1300			3-1/8" FROM CENTER	94000	58500	61500	30.0	59.3				57.5				
														3-1/8" FROM CENTER	84500	56700	58500	33.0	62.0				58.6				
														AVERAGE	90500	59150	59900	30.5	60.5				50.6				
														1" FROM CENTER	93000	56200	58800	29.0	54.1				54.4				
														3" FROM CENTER	94500	60000	60500	29.0	60.8				56.1				
											1500			5-1/8" FROM CENTER	97500	59000	60000	27.3	58.7	71300	28000	12.2		195 TO 207	41000	57000	.44	.28
														5-1/8" FROM CENTER	98000	62100	62500	30.0	63.0				53.6				
														AVERAGE	96250	59800	60400	28.8	59.6				54.7				
															100700	71000	75000	26.0	52.8					215				
										1 3/4"	1500	1			109000	72500	76500	26.5	54.8					218				
														AVERAGE	106850	72250	75750	26.2	54.8					216				
															142500	110000	112000	17.0	58.5					316				
											1000				138000	106000	106000	18.5	54.9					316	76000		.54	
VE	8442	1 1/2" DIA.	1600	1	1 1/2" DIA.	1550	1	OIL	1 1/4"			1		AVERAGE	139250	107500	109500	17.8	55.2					316				
															114700	93000	95000	25.0	56.1				53.6	241				
											1800				118000	93000	94000	22.0	54.7				52.0	243	58000	56000	.50	.31
														AVERAGE	116350	93000	94500	23.5	55.4				52.8	242				
															126500	95500	95600	21.0	53.3					248				
											1000				124700	92000	93500	22.0	52.8					239				
														AVERAGE	125600	93750	94550	21.5	53.0					250				
															113000	83800	83900	24.0	56.8					229				
										1" DIA.	1800	1 1/2			112500	84000	85000	24.5	57.5					229				
														AVERAGE	112750	83900	84450	24.2	57.1					229				
															108700	63800	63900	26.0	54.7					207				
VJ	8749	1" DIA.	1600	1 1/2							1500				106700	64000	65800	27.0	53.3					207				
														AVERAGE	108700	64900	65200	26.5	54.0					207				
															171000	155500	159400	10.0	58.5					334				
											1000				170000	152000	157000	10.0	56.7					330				
														AVERAGE	171000	153750	158200	10.0	56.1					340				
															121000	116500	117500	24.0	64.0					283				
					1" DIA.	1550	1 1/2	OIL	1" DIA.		1800	1 1/2			120500	122000	123500	24.0	62.0					282				
														AVERAGE	120500	119250	120000	24.0	63.0					282				

TABLE II (CONTINUED)

MECHANICAL AND ENDURANCE PROPERTIES OF NATIONAL EMERGENCY STEELS

MATERIAL DESIGNATION	NE DESIGNATION	HEAT TREATMENT: NORMALIZING SIZE OF SECTION	NORMALIZING TEMP DEG FAHR	NORMALIZING TIME HELD HOURS	QUENCHING SIZE OF TEST SPECIMEN	QUENCHING TEMP DEG FAHR	QUENCHING TIME HELD HOURS	QUENCHING COOLED	TEMPERING SIZE OF TEST SPECIMEN	TEMPERING TEMP DEG FAHR	TEMPERING TIME HELD HOURS	LOCATION AND DIRECTION OF TEST SPECIMENS	MECHANICAL PROPERTIES: TENSILE PROPERTIES TENSILE STRENGTH LB PER SQ INCH	PROOF STRESS LB PER SQ INCH	YIELD STRENGTH 1% SET LB PER SQ INCH	ELONGATION IN 2 INCHES PERCENT	REDUCTION OF AREA PERCENT	TORSIONAL PROPERTIES: ULTIMATE STRENGTH LB PER SQ INCH	PROP. LIMIT LB PER SQ INCH	MODULUS X10⁶ LB PER SQ INCH	CHARPY IMPACT FT/LB	BRINELL HARDNESS	ENDURANCE PROPERTIES: ENDURANCE LIMITS ROTATING CANTILEVER LB PER SQ INCH	ENDURANCE LIMITS ALTERNATING TORSION LB PER SQ INCH	ENDURANCE RATIOS: ENDURANCE LIMIT (FLEXURE) TS	ENDURANCE RATIOS: ENDURANCE LIMIT ALT TOR TS
WY	8749	6" DIA X 33" LONG	1600	7					QUADRANT 1½" LONG	1290	4	LONGITUDINAL 1/2" FROM CENTER	107900	69400	70000	26.0	58.9				27.6	217 TO 229					
												LONGITUDINAL 1-1/2" FROM CENTER	108700	72500	73500	25.0	56.2	78500	42000	11.9	27.6		49000	34000	.45	.31
												LONGITUDINAL AVERAGE	108100	70950	71750	25.5	56.0				27.6					
					1" SQ X 4-3/4"							TRANSVERSE 1-1/2 FROM OUTSIDE	107700	73500	74000	19.0	29.1									
									QUADRANT 1½" LONG	1300		LONGITUDINAL 1-1/2" FROM CENTER	98000	70200	70900	30.5	61.1				30.7					
												LONGITUDINAL 2-1/2" FROM CENTER	99000	66200	66800	30.0	61.1				29.0	207 TO 213				
												LONGITUDINAL 3-1/2" FROM CENTER	100200	70200	70700	28.0	61.1	73100	30000	12.0	29.6		44000	31000	.44	.31
												LONGITUDINAL AVERAGE	99100	68800	69400	29.5	61.1				29.7					
					1" SQ X 4-3/4"							TRANSVERSE 2-1/2 FROM OUTSIDE	98200	68000	64200	20.0	29.8									
XJ	8749	6" DIA X 36" LONG	1600	8					QUADRANT 1½" LONG	1200	6	LONGITUDINAL 1" FROM CENTER	104000	69500	70000	25.5	54.4				23.4					
												LONGITUDINAL 2" FROM CENTER	102700	70400	71000	25.0	58.6				28.3					
												LONGITUDINAL 3" FROM CENTER	100000	67500	68050	26.0	58.6	75200	38000	12.0	29.2	212 TO 218	47000	31000	.46	.31
												LONGITUDINAL 3-1/2" FROM CENTER	100000	67500	68000	27.5	60.2				28.8					
												LONGITUDINAL AVERAGE	101650	68700	69250	26.5	58.2				29.9					
					1" SQ X 4-3/4"							TRANSVERSE 1" FROM OUTSIDE	101000	68700	69200	23.0	42.2				13.7					
												TRANSVERSE 2" FROM OUTSIDE	101500	68700	69300	24.3	44.9				16.9					
												TRANSVERSE AVERAGE	101250	68700	69250	23.6	43.5				15.3					
									QUADRANT 1½" LONG	1300		LONGITUDINAL 1" FROM CENTER	99000	66500	67000	26.0	58.6				24.1					
												LONGITUDINAL 2" FROM CENTER	103500	70000	70500	24.0	54.7				26.3					
												LONGITUDINAL 3" FROM CENTER	100500	69000	69400	27.0	60.0	73600	36000	12.0	27.3	217 TO 226	46000	30000	.45	.30
												LONGITUDINAL 3-1/2" FROM CENTER	99000	68500	68900	32.0	60.0				29.8					
												LONGITUDINAL AVERAGE	101250	68000	68650	27.2	58.3				28.9					
					1" SQ X 4-3/4"							TRANSVERSE 1" FROM OUTSIDE	104500	70000	70400	18.5	41.8				16.7					
												TRANSVERSE 2" FROM OUTSIDE	109000	71200	71700	20.0	43.4				17.5					
												TRANSVERSE AVERAGE	107750	70600	71050	19.2	42.5				17.1					
WA	8749	1" DIA.	1600	1					1" DIA	1190	1		178500	135000		17.0	47.8									
													174200	138900	145000	17.5	47.1									
												AVERAGE	176350	136950	145000	17.2	47.4									
										1200			121500	119000	123000	18.0	58.5									
													121500	118700	122700	19.0	58.1									
												AVERAGE	121500	118350	122850	18.5	58.3									
										1325			134300	109000	103000	28.1	57.0									
													133300	99000	102500	21.2	57.3									
												AVERAGE	133800	99500	102750	24.7	57.1									
										1325			120700	76000	82500	23.9	58.1									
													121400	78500	82500	24.3	58.5									
												AVERAGE	121050	77250	82500	24.1	58.2									

WA	8749	1" DIA.	1500	1		1" DIA	1250	1		133500	99000	102500	21.2	57.3												
									AVERAGE	133000	99000	102750	20.7	57.1												
							1325			120700	78000	82500	23.9	58.1												
										121400	78500	82300	24.3	58.5												
									AVERAGE	121050	78250	82500	24.1	58.2												
							1400			134000	77500	104000	17.0	48.8												
										103200	79000	103000	17.5	47.6												
										102000	80500	106000	17.0	47.2												
									AVERAGE	113200	80250	104000	17.2	48.0												
VM	8749	6" DIA X 25" LONG	1425	8	QUADRANT 11" LONG		1000	3	LONGITUDINAL	1/2" FROM CENTER	148600	78050	101000	14.0	30.9	120100	41000	11.9	47	302 TO 311	53000	40000	.35	.27
										2" FROM CENTER	101100	78000	104000	16.0	34.8				46						
										3-1/4" FROM CENTER	149200	85000	108000	13.8	37.4				48						
										3-1/2" FROM CENTER	150700	87350	108000	16.7	36.8				63						
										AVERAGE	149300	82350	104750	15.5	35.5				51						
							1200			1/2" FROM CENTER	127100	61000	90000	16.4	43.3	88800	46500	12.0	143	258 TO 269	52000	33000	.42	.26
										2" FROM CENTER	122000	78000	86000	18.3	43.8										
										3-1/4" FROM CENTER	122500	80000	85500	20.5	47.4				136						
										3-1/2" FROM CENTER	128500	65000	94000	18.8	48.6				173						
										AVERAGE	125000	81000	85850	18.5	49.3				150						
							1300			1/2" FROM CENTER	106100	60500	65700	22.0	50.5	79500	41000	12.2	206	241 TO 248	51000	31000	.46	.28
										2" FROM CENTER	110300	91500	91500	22.6	48.6				212						
										3-1/4" FROM CENTER	110400	58000	66500	22.0	53.7				203						
										3-1/2" FROM CENTER	113400	74500	77000	22.3	52.7				234						
										AVERAGE	110050	61350	67650	22.2	51.4				213						
							1500			2"-C	146700	66000	94000	14.5	32.5	97800	29000	11.6	92	293 TO 311	52000		.38	
										3-1/4"-C	150000	57800	92500	14.5	31.4				114						
										3-1/2"-C	149000	58000	92000	14.0	32.6				102						
										3-1/4"-C	150000	63000	93000	14.0	32.5				112						
										AVG	148900	61100	92900	14.2	32.3				108						
WI	8749	6" DIA. X 24" LONG	1625	8	QUADRANT 11" LONG		1000	4	LONGITUDINAL	3/4" FROM CENTER	126000	81500	87500	20.5	45.0	88300	19000	11.7	161	269 TO 277	55000	37000	.45	.31
										2" FROM CENTER	123800	84000	87500	20.0	43.4										
										3" FROM CENTER	125900	85000	88500	21.0	46.0				130						
										3-1/4" FROM CENTER	107500	61000	61500	21.5	53.1				154						
										AVERAGE	121000	77250	81250	20.7	42.0				148						
							1200			3/4" FROM CENTER	104700	71000	71500	22.0	52.5	75600	19000	12.0	226	223 TO 229	48000	33000	.45	.31
										1-3/4" FROM CENTER	108200	71000	71500	24.0	53.8				233						
										2" FROM CENTER	107000	71200	71500	24.5	53.0				204						
										3" FROM CENTER	105800	69000	70000	25.5	55.7										
										3-1/2" FROM CENTER	106800	71600	72000	25.0	54.8										
										AVERAGE	105800	70750	71300	23.7	54.8				222						
							1500			3/4" FROM CENTER	94000	58700	59200	30.0	61.8	72500	19000	11.8	250	197 TO 204	42000	28000	.44	.30
										1-3/4" FROM CENTER	96800	61500	65000	27.5	59.8				259						
										2" FROM CENTER	96500	62500	63000	26.5	60.0				261						
										3-1/2" FROM CENTER	95700	58500	60000	29.0	61.0				252						
										3-1/2" FROM CENTER	96800	62000	62500	29.0	60.8										
										AVERAGE	95900	60650	61950	28.4	60.8				277						

		SIZE OF SECTION	TEMP DEG FAHR	TIME HELD HOURS	SIZE OF INSERT	TEMP DEG FAHR	TIME HELD HOURS	COOLED	SIZE OF INSERT	TEMP DEG FAHR	TIME HELD HOURS	TEST SPECIMENS	STRENGTH	STRESS	STRENGTH 1% SET	IN 2 INCHES	OF AREA	STRENGTH	LIMIT	MODULUS X10⁶	IMPACT FT/LB	BRINELL HARDNESS	NOTCHED CANTILEVER	ALTERNATING TORSION	ENDURANCE LIMIT (FLEXURE)	ENDURANCE LIMIT AXIAL TEN
													LB PER SQ INCH			PERCENT		LB PER SQ INCH					LB PER SQ INCH		TS	TS
WY	8749	7" DIA X 23" LONG	1600	7					QUADRANT 11" LONG	1200	4	1/2" FROM CENTER	107500	69400	70000	26.0	55.9	73500	42000	11.9	27.6	217 TO 229	49000	34000	.45	.31
												1-1/2" FROM CENTER	108700	72500	73500	25.0	56.2				27.6					
												AVERAGE	108100	70950	71750	25.5	56.0				27.6					
									1" SQ X 6-3/4"			TRANSVERSE 1-1/2 FROM OUTSIDE	107700	73800	74000	18.0	29.1									
									QUADRANT 11" LONG	1200	4	1-1/2" FROM CENTER	96000	70900	70700	30.5	61.1	73100	39000	12.0	30.7	207 TO 213	44000	31000	.44	.31
												2-1/2" FROM CENTER	99000	68200	68800	30.0	61.1				29.0					
												3-1/2" FROM CENTER	100200	70200	70700	28.0	61.1				29.6					
												AVERAGE	98100	69800	69400	29.5	61.1				28.7					
									1" SQ X 6-3/4"			TRANSVERSE 2-1/2 FROM OUTSIDE	98200	62000	64200	20.0	29.8									
XJ	8749	6" DIA X 24" LONG	1600	8					QUADRANT 11" LONG	1200	4	1" FROM CENTER	104000	69500	70000	25.5	54.4	72200	38000	12.0	23.4	212 TO 210	47000	31000	.46	.31
												2" FROM CENTER	102700	70450	71000	25.0	56.6				28.3					
												3" FROM CENTER	100000	67500	68000	28.0	56.6				28.2					
												3-1/2" FROM CENTER	100000	67500	68000	27.5	62.2				26.0					
												AVERAGE	101850	68750	69250	26.5	58.2				26.9					
									1" SQ X 6-3/4"			1" FROM OUTSIDE	101000	68700	69200	23.0	42.8				13.7					
												1" FROM OUTSIDE	101500	68700	69300	24.3	44.3				14.9					
												AVERAGE	101250	68700	69250	23.6	43.5				13.3					
									QUADRANT 11" LONG	1200	4	1" FROM CENTER	98000	66500	67000	26.0	58.6	75600	38000	12.0	24.1	217 TO 229	46000	30000	.45	.30
												2" FROM CENTER	105500	70000	70500	24.0	54.7				28.3					
												3" FROM CENTER	100300	69000	69400	27.0	60.0				27.3					
												3-1/2" FROM CENTER	98000	66500	66900	32.0	60.0				28.6					
												AVERAGE	101250	69000	69450	27.2	56.3				26.9					
									1" SQ X 6-3/4"			1" FROM OUTSIDE	106300	70000	70400	18.5	41.6				16.7					
												1" FROM OUTSIDE	108000	71200	71700	20.0	43.4				17.8					
												AVERAGE	107750	70600	71050	19.2	42.5				17.1					
WA	8749	1" DIA	1600	1					1" DIA	1250	1		178500	135000		17.0	47.8									
										1100			174800	135900	145000	17.5	47.1									
												AVERAGE	176350	136450	145000	17.2	47.4									
										1300			121500	118000	123000	18.0	35.5									
													121500	118700	122700	19.0	35.1									
												AVERAGE	121500	118350	122850	18.5	35.3									
													154300	100000	103000	20.1	57.0									
													133300	98000	102500	21.2	57.3									
												AVERAGE	133800	98500	102750	20.7	57.1									
										1325			120700	76000	82500	23.0	38.1									
													121400	75500	82500	24.3	58.3									
												AVERAGE	121050	75750	82550	24.1	58.2									
										1400			104800	77500	104000	17.0	45.8									
													103200	79000	103000	17.5	47.6									
													102000	80300	105000	17.0	47.2									
												AVERAGE	103300	80900	104000	17.2	46.9									
										1000		1/2" FROM CENTER	148850	79000	101500	14.0	36.9	100000	41000	11.9	47	302 TO 311	53000	40000	.35	.27
												2" FROM CENTER	151100	78000	104000	16.0	34.8				46					
												2-1/2" FROM CENTER	148200	86000	108000	15.5	37.0				48					
												3-1/2" FROM CENTER	150700	87000	108000	16.7	38.8				43					
												AVERAGE	149700	82250	105375	15.6	36.9				46					

										AVERAGE	153500	89000	104000	17.2	469													
VM	8749	6" DIA. X 26" LONG	1625	8		QUADRANT 11" LONG	1000	3	LONGITUDINAL	1/2" FROM CENTER	148800	78000	101000	14.0	363	40000	41000	11.9	47	302 TO 311	53000	40000	.25	.27
										2" FROM CENTER	151100	76000	104000	14.0	348				46							
										2-1/4" FROM CENTER	148200	85000	108000	15.8	374				48							
										2-1/2" FROM CENTER	150700	87500	108000	16.7	388				63							
										AVERAGE	149800	82350	104750	15.6	355				91							
							1200			1/2" FROM CENTER	127100	81000	98000	16.4	435	38800	46500	12.0	143	255 TO 269	52000	33000	.42	.56
										2" FROM CENTER	122000	78000	94000	19.3	458											
										2-1/2" FROM CENTER	122500	80000	88500	20.5	474				136							
										2-1/2" FROM CENTER	128500	85300	94000	18.0	426				172							
										AVERAGE	123000	81000	88850	18.5	403				180							
							1300			1/2" FROM CENTER	108100	68500	85700	22.0	505	79900	41000	12.2	206	241 TO 248	51000	31000	.44	.58
										2" FROM CENTER	110300	71500	81500	22.6	486				212							
										2-1/4" FROM CENTER	110400	59000	84500	22.0	537				203							
										2-1/2" FROM CENTER	113400	74500	77000	22.3	527				234							
										AVERAGE	110080	61350	87650	22.2	514				213							
							1500			2"-C	146700	66000	94000	14.5	328	57800	29000	11.8	92	293 TO 311	52000		.38	
										2-1/4"-C	150000	57500	92500	14.5	314				114							
										2-1/2"-C	149000	58000	92000	14.0	328				102							
										2-1/4"-C	150000	63000	93000	14.0	329				112							
										AVG	148900	61100	92900	14.2	323				108							
WI	8749	6" DIA. X 26" LONG	1625	8		QUADRANT 11" LONG	1000	4	LONGITUDINAL	3/4" FROM CENTER	126000	81500	97000	20.5	450	84100	19000	11.7	161	260 TO 277	55000	37000	.45	.51
										2" FROM CENTER	123300	84000	97500	20.0	434											
										2-1/4" FROM CENTER	129000	85000	98500	21.0	460				150							
										2-1/2" FROM CENTER	107500	61000	81500	21.5	331				154							
										AVERAGE	121000	77250	91250	20.7	420				148							
							1300			3/4" FROM CENTER	104700	71000	71500	28.0	525	75600	19000	12.0	226	285 TO 229	45000	33000	.45	.51
										1-3/4" FROM CENTER	106200	71000	71500	24.0	538				233							
										2" FROM CENTER	107000	71200	71500	24.5	520				206							
										2" FROM CENTER	108200	68000	70000	24.5	557											
										2-1/2" FROM CENTER	106900	71600	72000	25.5	548											
										AVERAGE	106900	70750	71300	25.7	548				222							
							1500			3/4" FROM CENTER	94000	58700	59200	30.0	618	72300	19000	11.8	258	197 TO 204	42000	28000	.44	.30
										2-1/4" FROM CENTER	96500	61500	65000	27.5	586				298							
										2-1/2" FROM CENTER	96000	62500	63000	26.5	600				281							
										2-1/2" FROM CENTER	93700	58500	60000	29.0	616				292							
										2-1/2" FROM CENTER	96800	62000	62500	29.0	606											
										AVERAGE	95900	60650	61950	28.4	608				277							
							1800			3/4" FROM CENTER	111500	56500	55000	21.0	368	80400	17700	12.1	136	224 TO 231	46000		.40	
										2-1/4" FROM CENTER	120500	56500	56000	24.0	417				154							
										2-1/2" FROM CENTER	109500	56000	57500	23.0	421				161							
										2-1/2" FROM CENTER	120500	56900	57000	25.0	466				227							
										2-1/2" FROM CENTER	110500	55400	56000	24.5	421											
										AVERAGE	114400	56000	56400	23.7	417				169							

TABLE II (CONTINUED)

MECHANICAL AND ENDURANCE PROPERTIES OF NATIONAL EMERGENCY STEELS

MATERIAL DESIGNATION	NE DESIGNATION	HEAT TREATMENT										LOCATION AND DIRECTION OF TEST SPECIMENS	MECHANICAL PROPERTIES									ENDURANCE PROPERTIES			
		NORMALIZING			QUENCHING				TEMPERING				TENSILE PROPERTIES					TORSIONAL PROPERTIES			CHARPY IMPACT FT/LB	CHARPY HARDNESS	ENDURANCE LIMITS		ENDURANCE RATIOS	
		SIZE OF SECTION	TEMP DEG. FAHR.	TIME HELD HOURS	SIZE OF SECT. INCHES	TEMP DEG. FAHR	TIME HELD HOURS	COOLED	SIZE OF SECT. INCHES	TEMP DEG. FAHR	TIME HELD HOURS		TENSILE STRENGTH	PROOF STRESS	YIELD STRENGTH .1% SET	ELONGATION IN 2 INCHES	REDUCTION OF AREA	ULTIMATE STRENGTH	PROP. LIMIT	MODULUS 2104			ROTATING CANTILEVER	ALTERNATING TORSION	ENDURANCE LIMIT (FLEXURE) / T.S.	ENDURANCE LIMIT ALT TOR / T.S.
													LB PER SQ INCH			PERCENT		LB PER SQ INCH					LB PER SQ INCH			
XQ	8749	1" DIA.	1600	1						1050			123700	89100	91500	19.0	48.1									
													126000	92800	95600	18.5	47.6									
													128000	91200	91500	20.0	47.5									
													126000	91200	91300	19.0	47.1									
												AVERAGE	126250	91050	92600	19.1	47.5									
										1200			108700	81900	81600	21.0	53.0									
													111300	81300	82000	23.0	52.5									
									1" DIA		1		114000	81200	81500	23.0	53.0									
													113700	82500	82700	23.0	53.5									
												AVERAGE	111950	81850	82000	23.0	53.0									
										1300			106300	80000	80300	21.5	48.1									
													99200	72550	73050	24.0	56.8									
												AVERAGE	102750	76250	76750	22.7	52.5									
										1475			101500	63900	62500	29.0	54.3									
													106000	60600	61000	27.0	53.8									
												AVERAGE	103750	62250	63250	28.0	54.0									
										1050			180300	165000	170200	15.0	46.2									
													180300	165000	166200	14.0	44.0									
												AVERAGE	180300	165000	168200	14.5	45.1									
										1200			136000	125600	126000	20.5	58.9									
													142300	133000	133500	20.0	54.7									
													137700	123700	124000	21.0	58.7									
													134500	119700	120000	21.5	57.3									
					1" DIA	1850	1	OIL	1" DIA.		1	AVERAGE	138200	125500	125900	20.8	58.8									
										1300			124000	116100	116700	23.0	56.5									
													120800	116200	116700	23.5	57.7									
													116000	105400	106000	25.5	60.7									
													115500	105400	106000	26.0	59.8									
												AVERAGE	119000	111250	111850	25.0	59.2									
										1100			128500	100500	100800	23.0	52.7									
													128200	102000	102500	21.5	51.6									
												AVERAGE	128350	101250	101650	22.2	52.1									
										1200			120800	93500	93700	21.0	53.2									
													119500	92600	93500	22.5	53.5									
									1" DIA.		1	AVERAGE	120000	93200	93600	21.7	54.3									
													113000	87500	87900	23.5	54.9									

| | | |

TABLE II (CONTINUED)

MECHANICAL AND ENDURANCE PROPERTIES OF NATIONAL EMERGENCY STEELS

MATERIAL DESIGNATION	NE DESIGNATION	HEAT TREATMENT										LOCATION AND DIRECTION OF TEST SPECIMENS	MECHANICAL PROPERTIES									ENDURANCE PROPERTIES			
		NORMALIZING			QUENCHING				TEMPERING				TENSILE PROPERTIES					TORSIONAL PROPERTIES			CHARPY IMPACT FT/LB	BRINELL HARDNESS	ENDURANCE LIMITS		ENDURANCE RATIOS	
		SIZE OF SECTION	TEMP DEG. FAHR.	TIME HELD HOURS	SIZE OF TEST SPECIMEN	TEMP DEG FAHR	TIME HELD HOURS	COOLED	SIZE OF TEST SPECIMEN	TEMP DEG. FAHR.	TIME HELD HOURS		TENSILE STRENGTH	PROOF STRESS	YIELD STRENGTH 1/2% SET	ELONGATION IN 2 INCHES	REDUCTION OF AREA	ULTIMATE STRENGTH	PROP LIMIT	MODULUS X10^6			ROTATING CANTILEVER	ALTERNATING TORSION	ENDURANCE LIMIT (FLEXURE) / TS	ENDURANCE LIMIT ALT TOR / TS
													LB PER SQ INCH			PERCENT		LB PER SQ INCH					LB PER SQ INCH			
XE	8949	6" DIA X 24" LONG	1600	6					QUADRANT 11" LONG	1300	4	LONGITUDINAL 2" FROM CENTER	97000	62300	63000	27.5	57.2	72000	34400	12.0	27.6	201 TO 212	48000	27500	.43	.28
												LONGITUDINAL 3" FROM CENTER	97500	61200	64800	29.5	57.7				24.0						
												LONGITUDINAL 3-1/8" FROM CENTER	96800	59000	62500	27.5	57.5				26.0						
												LONGITUDINAL AVERAGE	97100	60800	63400	28.1	57.5				26.1						
									1" SQ X 4-3/4"			TRANSVERSE 1" FROM OUTSIDE	96000	61200	61700	23.5	34.4				15.4						
												TRANSVERSE 1" FROM OUTSIDE	98000	63700	64300	27.0	44.8				14.6						
												TRANSVERSE AVERAGE	96500	62500	62950	25.2	39.5				15.0						
									QUADRANT 11" LONG	1300	4	LONGITUDINAL 1" FROM CENTER	94000	59700	60400	29.5	58.9	74800	33000	12.0	29.7	197 TO 203	43500	30000	.46	.32
												LONGITUDINAL 2" FROM CENTER	96000	59500	60000	30.0	60.2				13.0						
												LONGITUDINAL 3" FROM CENTER	94000	58700	59400	29.5	58.6				29.0						
												LONGITUDINAL 3-1/8" FROM CENTER	95800	61200	62500	29.5	59.3				27.6						
												LONGITUDINAL AVERAGE	94900	59800	60400	29.6	59.2				25.5						
									1" SQ X 4-3/4"			TRANSVERSE 1" FROM OUTSIDE	101600	65000	65400	23.0	41.5				18.0						
												TRANSVERSE 1" FROM OUTSIDE	99000	63200	63700	22.0	34.1				17.8						
												TRANSVERSE AVERAGE	100250	64100	64650	22.5	37.8				17.9						
VZ	8949	1" DIA.	1600	1					1" DIA.	1100	1		130500	89000	95500	20.5	52.2										
													132000	86500	96500	19.5	50.8										
												AVERAGE	131250	87750	96000	20.0	51.5										
										1200			123500	91000	91500	20.0	54.9										
													123200	88500	92500	21.0	54.8										
												AVERAGE	123350	89750	91750	20.5	54.8										
										1300			111500	82000	84000	22.0	59.3										
													108500	79000	80500	24.0	60.5										
												AVERAGE	110000	80000	82250	23.0	59.8										
										1450			101100	61500	62000	28.5	57.7										
													101700	61500	62300	29.0	58.9										
												AVERAGE	101400	61500	62290	28.7	57.5										
					1" DIA	1850	1	OIL	1" DIA.	1200	1		146500	130000	131000	20.0	58.2										
													144000	129000	130500	21.0	58.0										
												AVERAGE	145250	129750	130750	20.5	58.8										
										1200		8" FROM CENTER	102400	71000	72500	22.5	48.1	73600	27000	11.8	26.6	221 TO 232	51000	33000	.49	.32
												3-1/2" FROM CENTER	104000	72500	75200	23.0	51.1										
												3" FROM CENTER	106500	78100	78700	24.0	51.4				22.6						
												3-1/8" FROM CENTER	105500	75800	79400	24.5	50.8				21.8						
												AVERAGE	104600	75100	76700	23.8	50.3				23.3						
												1/8" FROM CENTER	90600	61000	62500	26.5	54.7				33.6						
												2" FROM															

VY	8949	8" DIA. X 23" LONG	1625	8		QUADRANT 11" LONG	1200	4	LONGITUDINAL	3" FROM CENTER	104000	72900	75200	23.0	51.1	73600	27000	11.8	22.6	221 TO 232	51000	35000	49	32
										3-1/4" FROM CENTER	105500	76800	79400	24.5	50.8				21.8					
										AVERAGE	104600	75100	76700	23.8	50.3				23.3					
							1300			1/2" FROM CENTER	90600	61000	62900	28.5	54.7	72300	25800	11.7	33.6	197 TO 203	47000	28000	50	31
										2" FROM CENTER	92200	65600	66000	28.5	60.1				27.7					
										3-1/4" FROM CENTER	96000	67300	68000	27.5	59.3				29.0					
										3-1/2" FROM CENTER	94000	66000	66200	28.5	60.0				27.9					
										AVERAGE	93480	65000	65700	27.7	58.5				29.3					
							1475			1/2" FROM CENTER	93200	56800	57500	27.5	51.4	72400	26000	11.9	20.6	198 TO 215	44000	29000	.44	29
										2" FROM CENTER	97000	57500	58000	23.3	41.5				19.9					
										3-1/4" FROM CENTER	103500	61500	62300	26.0	43.4				20.0					
										3-1/2" FROM CENTER	102000	62500	62100	26.0	44.9				18.9					
										3-1/2" FROM CENTER	104500	55000	60000	24.5	44.9									
										AVERAGE	100000	58300	60200	25.5	45.2				20.1					
VN	8949	1" DIA	1600	1		1" DIA	1100	1			138500	96000	101000	19.0	52.7					302				
											158100	99000	103000	22.2	55.1					302				
										AVERAGE	157300	95500	102000	20.2	53.3					302				
							1300				120600	89000	90000	22.8	57.9					285				
											120900	88000	91000	22.0	57.4					285				
										AVERAGE	120800	88500	90500	22.4	57.6					285				
							1450				108200	79000	80000	23.0	60.9					207				
											108200	80000	80200	23.2	62.5					207				
										AVERAGE	108200	79500	80100	23.1	61.7					207				
											116500	75000	75200	24.8	56.1									
											114300	73000	75000	24.0	56.5									
										AVERAGE	115400	74000	75100	24.4	56.3									
					1" DIA.	1550	1	OIL	1" DIA.	1200	1			129000	131000	131000	20.3	56.9								
														129800	131000	131500	20.3	55.8							
										AVERAGE				129400	131000	131500	20.3	56.3							
		AS RECEIVED 13-3/4" DIA.							LONGITUDINAL	3" FROM CENTER	106700	43000	44200	18.0	24.8					217 TO 225	42000	27000	39	.25
										3-1/4" FROM CENTER	106800	42000	44900	18.0	24.1									
										AVERAGE	106850	42500	44100	18.5	24.4									
		AS RECEIVED 13-3/4" DIA.				QUADRANT 11" LONG	1100	3	LONGITUDINAL	3-1/4" FROM CENTER	105800		45000	19.0	23.4	76800	21600	12.2		212 TO 216	42000	27000	.41	26
										3-1/4" FROM CENTER	104100	42500	43200	18.5	24.4	77000	21800	12.4					
										3" FROM CENTER	102100	43800	44500	20.0	28.9									
										6" FROM CENTER	102900	38800	44800	16.8	24.5									
										AVERAGE	103650	41700	44850	18.8	25.3	76800	21600	12.3						
									TRANSVERSE	3-1/2" FROM CENTER	103300	43000	46000	9.5	7.4									
										3-1/2" FROM CENTER	98900	41500	42500	9.0	8.1									
										3-1/2" FROM CENTER	103100	42000	43500	11.0	8.9									
										AVERAGE	101750	42150	44100	9.8	8.1									
						QUADRANT 11" LONG			LONGITUDINAL	3-1/4" FROM CENTER	105400	41800	46000	20.0	23.1					217 TO				
										3-1/4" FROM CENTER	105100	42500	45800	18.5	25.2									
										6" FROM CENTER	102300	42800	45400	20.0	27.2									

												1400			114300	73000	78000	25.0	96.5											
													AVERAGE		115400	74000	78100	25.4	96.3											
						1" DIA.	1980	1	OIL	1" DIA.	1200	1			129000	131000	131000	20.3	96.9											
															129800	131000	131500	20.3	95.8											
													AVERAGE		129400	131000	131500	20.3	96.3											
VD	8949	AS RECEIVED 13-3/4" DIA.											LONGITUDINAL	3" FROM CENTER	106700	43500	46200	18.0	24.8				217 TO 223	42000	27000	.39	.25
														3" FROM CENTER	105600	42000	46000	18.0	24.1											
														AVERAGE	106650	42500	46100	18.0	24.4											
		AS RECEIVED 13-3/4" DIA.				QUADRANT 11" LONG		1100	3				LONGITUDINAL	3-1/2" FROM CENTER	106400		45000	19.0	25.4	76900	21600	12.5	218 TO 216	42000	27000	.41	.26
														3-1/2" FROM CENTER	104100	42500	45200	18.5	24.6	77000	21600	12.4					
														6" FROM CENTER	102100	43000	46500	20.0	25.9								
														6" FROM CENTER	102900	38000	46500	18.0	24.5								
														AVERAGE	103650	41700	45800	19.0	25.5	76900	21600	12.3					
													TRANSVERSE	3-1/2" FROM CENTER	103300	43000	46000	9.5	7.6								
														3-1/2" FROM CENTER	98900	41500	42500	9.0	8.1								
														3-1/2" FROM CENTER	103100	42000	43500	11.0	8.9								
														AVERAGE	101750	42150	44100	9.8	8.1								
								1800	2				LONGITUDINAL	3-1/2" FROM CENTER	108400	42500	46000	20.0	23.1				217 TO 224	38000	27500	.37	.27
														3-1/2" FROM CENTER	105100	42500	45900	18.5	25.2								
														6" FROM CENTER	102300	42800	46400	20.0	27.2								
														6" FROM CENTER	101900	43000	46400	19.5	26.5								
														AVERAGE	103700	42750	46050	19.5	25.5								
													TRANSVERSE	3-1/2" FROM CENTER	102100	42500	46500	10.0	10.0								
														3-1/2" FROM CENTER	101100	42500	46500	11.0	11.3								
														3-1/2" FROM CENTER	104000	43000	46000	10.0	8.9								
														AVERAGE	102400	42750	46250	10.3	10.1								
		1" DIA	1600	6		1" DIA	1300	2					LONGITUDINAL		134200	105500	108000	17.5	53.3								
															127000	99500	100050	21.0	57.5								
														AVERAGE	130600	102500	104500	19.2	55.4								
		QUADRANT 11" LONG	1600	6		QUADRANT 11" LONG	1300	6							107000	72000	75000	24.5	60.5	78400	37500	12.1					
															105000	66000	70000	25.0	60.1	79100	39200	12.1	223 TO 230	50000	31000	.47	.29
															105800	69000	73000	25.0	60.6								
															107700	72000	75000	23.0	58.5								
														AVERAGE	106300	69750	73250	24.0	59.9	78750	39350	12.1					
													RADIAL		104000	67500	71000	17.5	25.0								
															104500	71000	74000	15.0	21.0								
															105700	72500	77500	11.5	15.4								
														AVERAGE	105700	71250	76150	14.7	20.1								

TABLE II (CONTINUED)

MECHANICAL AND ENDURANCE PROPERTIES OF NATIONAL EMERGENCY STEELS

MATERIAL DESIGNATION	NE DESIGNATION	HEAT TREATMENT: NORMALIZING: SIZE OF SECTION	NORMALIZING: TEMP DEG FAHR.	NORMALIZING: TIME HELD HOURS	QUENCHING: SIZE OF TEST SPECIMEN	QUENCHING: TEMP DEG FAHR	QUENCHING: TIME HELD HOURS	QUENCHING: COOLED	TEMPERING: SIZE OF TEST SPECIMEN	TEMPERING: TEMP DEG. FAHR	TEMPERING: TIME HELD HOURS	LOCATION AND DIRECTION OF TEST SPECIMENS	MECHANICAL PROPERTIES: TENSILE PROPERTIES: TENSILE STRENGTH (LB PER SQ INCH)	PROOF STRESS (LB PER SQ INCH)	YIELD STRENGTH 1% SET (LB PER SQ INCH)	ELONGATION IN 2 INCHES (PERCENT)	REDUCTION OF AREA (PERCENT)	TORSIONAL PROPERTIES: ULTIMATE STRENGTH (LB PER SQ INCH)	TORSIONAL PROPERTIES: PROP LIMIT (LB PER SQ INCH)	TORSIONAL PROPERTIES: MODULUS KIPS (LB PER SQ INCH)	CHARPY IMPACT FT/LB	BRINELL HARDNESS	ENDURANCE PROPERTIES: ENDURANCE LIMITS: ROTATING CANTILEVER (LB PER SQ INCH)	ENDURANCE LIMITS: ALTERNATING TORSION (LB PER SQ INCH)	ENDURANCE RATIO: ENDURANCE LIMIT FLEXURE / T.S.	ENDURANCE RATIO: ENDURANCE LIMIT ALT TOR / T.S.
VD (CONTINUED)	8949				(1-3/4" X 11" LONG)	1600	6	OIL	QUADRANT 11" LONG	1100	12	LONGITUDINAL 1" FROM CENTER	173000	123100	143600	11.0	30.9					341 TO 364				
												LONGITUDINAL 1" FROM CENTER	174900	120700	147000	15.0	38.0									
												LONGITUDINAL 3-1/4" FROM CENTER	168100	124800	138000	16.0	40.6	108900	70000	12.0						
												LONGITUDINAL 3-1/4" FROM CENTER	164000	123000	138800	16.0	44.5									
												LONGITUDINAL 6" FROM CENTER	170900	130900	145400	14.5	41.2	109400	70000	12.0						
												LONGITUDINAL 6" FROM CENTER	169400	133700	146800	17.0	43.6	110000	69000	12.0						
												LONGITUDINAL AVERAGE	170450	125950	142150	14.9	38.1	102450	69650	12.0						
												RADIAL 2" FROM CENTER	156700	120000		3.0	12.5									
												RADIAL 2" FROM CENTER	163400	122600	133700	7.0	7.5									
												RADIAL 2" FROM CENTER	165600	128600	139800	8.0	6.2									
												RADIAL AVERAGE	161900	127000	136750	5.0	8.5									
										1200	4	LONGITUDINAL 1" FROM CENTER	125300	98800	103400	6.0	14.4									
												LONGITUDINAL 1" FROM CENTER	134700	102900	108100	14.0	27.5									
												LONGITUDINAL 3-1/4" FROM CENTER	130800	100300	102300	20.0	52.7	83900	54000	12.0						
												LONGITUDINAL 3-1/4" FROM CENTER	133000	101200	105000	17.0	48.3					262 TO 285				
												LONGITUDINAL 6" FROM CENTER	127800	100300	101800	21.0	55.5	83900	54000	12.1						
												LONGITUDINAL 6" FROM CENTER	130700	102100	104000	19.0	56.0	83000	56000	12.0						
												LONGITUDINAL AVERAGE	130400	100950	104150	16.1	42.9	83250	55350	12.0						
												RADIAL 2" FROM CENTER	128600	101200	103200	5.5	14.8									
												RADIAL 2" FROM CENTER	128300	99400	101200	8.0	8.5									
												RADIAL 2" FROM CENTER	123700	104300	107700	8.0	8.9									
												RADIAL AVERAGE	126850	101650	104050	8.5	10.0									
										1200	4	LONGITUDINAL 1" FROM CENTER	122300	89500	93500	22.0	56.2									
												LONGITUDINAL 1" FROM CENTER	128300	93200	96700	20.0	53.5									
												LONGITUDINAL 3-1/4" FROM CENTER	116800	87000	94500	23.0	55.9	80700	50000	12.0						
												LONGITUDINAL 3-1/4" FROM CENTER	122000	90000	92300	19.0	56.0					252 TO 257				
												LONGITUDINAL 6" FROM CENTER	119300	91100	92900	21.0	57.3	79700	52000	11.9						
												LONGITUDINAL 6" FROM CENTER	121800	94500	94700	22.0	56.2	79100	52000	12.0						
												LONGITUDINAL AVERAGE	121000	90000	93000	21.1	55.8	79850	51300	12.0						
												RADIAL 2" FROM CENTER	119300	89500	91700	13.0	16.7									
												RADIAL 2" FROM CENTER	122600	90700	96000	10.0	13.4									
												RADIAL 2" FROM CENTER	118300	87000	89500	12.0	33.4									
												RADIAL AVERAGE	120100	89050	92400	11.7	21.2									
										1200	4	LONGITUDINAL 1" FROM CENTER	116500	88500	88000	25.0	57.3									
												LONGITUDINAL 1" FROM CENTER	114500	83700	86200	23.0	47.8									
												LONGITUDINAL 3-1/4" FROM CENTER	110900	78300	88900	25.5	56.8	78700	44000	12.0						
												LONGITUDINAL 3-1/4" FROM CENTER	113300	80200	81400	22.0	56.2					229 TO 255				
												LONGITUDINAL 6" FROM CENTER	110000	80000	82000	24.0	57.0	77000	44000	12.0						
												LONGITUDINAL 6" FROM CENTER	112000	82000	84000	24.0	57.0	77000	44000	12.0						
												LONGITUDINAL AVERAGE	111150	80200	83400	24.0	57.0	77850	44000	12.0						
												RADIAL 2" FROM CENTER	113000	84000	86000	10.0	13.4									
												RADIAL 2" FROM CENTER	113000	84000	86000	10.0	13.4									
												RADIAL 2" FROM CENTER	113000	84000	86000	10.0	13.4									
												RADIAL AVERAGE	113000	84000	86000	10.0	13.4									

													AVERAGE	120100	89050	92400	11.7	21.2							
													1" FROM CENTER	118500	86500	89000	25.0	37.3							
													2" FROM CENTER	114500	83700	86200	23.0	47.8							
													3-1/8" FROM CENTER	110900	78300	86500	25.5	58.9	76700	44000	12.0				
													3-1/8" FROM CENTER	113300	80200	81400	22.0	56.2							
													4" FROM CENTER	110900	76300	80500	23.0	58.8	79700	43200	12.0				
													5" FROM CENTER	112900	84200	88200	23.0	59.5	82000	45000	12.0				
													AVERAGE	113150	81500	85100	23.6	56.7	79450	44090	12.0				
													2" FROM CENTER	114000	82500	86000	16.0	21.3							
													5" FROM CENTER	108900	76300	79800	13.5	18.5							
													8" FROM CENTER	111900	77600	82000	15.0	20.2							
													AVERAGE	111600	78500	82150	14.8	20.3							
XL	9445	8" DIA. X 36" LONG	1650	8			QUADRANT 11" LONG	1200	4	LONGITUDINAL	1" FROM CENTER	113000	84200	85000	22.0	47.8				23.4	207 TO 212	48000	31000	.42	.29
											2" FROM CENTER	103700	87200	87700	25.5	54.3									
											3" FROM CENTER	106700	82500	83000	24.0	58.5	73500	38000	12.1						
											3-1/8" FROM CENTER	102000	89400	90000	28.5	57.3				24.7					
											AVERAGE	106350	80800	81400	25.2	53.7				24.0					
							1" SQ. X 4-3/4"			TRANSVERSE	1" FROM OUTSIDE	125800	84000	88000	21.0	48.8				16.2					
											2" FROM OUTSIDE	111000	92200	92500	17.0	28.8				15.4					
											AVERAGE	118250	73080	75750	19.0	37.7				14.8					
							QUADRANT 11" LONG	1200	4	LONGITUDINAL	1" FROM CENTER	101500	96000	98500	27.0	54.8				21.9					
											2" FROM CENTER	100000	95500	97500	27.0	53.3				24.7					
											3" FROM CENTER	98000	96000	98000	30.0	58.8	71100	28400	12.0	25.7	197 TO 217	43000		.43	
											3-1/8" FROM CENTER	99500	81000	81300	29.0	58.6				26.1					
											AVERAGE	99750	97850	99300	28.2	56.3				24.3					
							1" SQ. X 4-3/4"			TRANSVERSE	1" FROM OUTSIDE	99000	98000	98400	21.8	36.7				14.9					
											1" FROM OUTSIDE	99500	97500	97700	24.0	41.6				14.9					
											AVERAGE	99250	98250	98350	22.7	38.6				14.9					
XI	9445	10" DIA. X 36" LONG	1850	10			QUADRANT 11" LONG	1200	8	LONGITUDINAL	1" FROM CENTER	116500	84000	86700	18.5	27.5				10.7					
											2" FROM CENTER	117500	86200	88500	22.0	40.4				18.6					
											3" FROM CENTER	116500	84400	86500	22.0	39.6				14.2					
											3-1/8" FROM CENTER	115500	87500	88100	24.0	43.1	63400	26000	12.0	14.2	235 TO 241	47000	32000	.40	.28
											4-1/2" FROM CENTER	116500	89200	90000	22.0	46.5				18.7					
											AVERAGE	116500	86250	88300	21.7	39.6				14.5					
							1" SQ. X 4-3/4"			TRANSVERSE	1" FROM OUTSIDE	118000	97500	97700	20.0	32.1				11.9					
											2" FROM OUTSIDE	114000	93100	94100	17.5	23.7				12.8					
											AVERAGE	116250	95300	95900	18.7	27.9				12.3					
							QUADRANT 11" LONG	1200		LONGITUDINAL	1" FROM CENTER	116300	48000	95700	18.5	38.7				16.7					
											2" FROM CENTER	121300	80600	82300	19.5	43.6				18.2	237 TO 241	48000		.40	
											3-1/8" FROM CENTER	122000	63000	63700	20.0	48.1	65100	26200	12.0	18.8					
											AVERAGE	119900	97200	80580	19.3	42.6				16.9					
							1" SQ. X 4-3/4"			TRANSVERSE	1" FROM OUTSIDE	120000	97500	99000	13.0	13.4									
											1" FROM OUTSIDE	118000	94200	97500	15.5	21.3									
											AVERAGE	119250	95850	98250	14.2	17.3									
		9" DIA.									1" FROM			93700	22.0	39.6									

VD (CONTINUED)	8949			12-3/4" X 11" LONG	1200	6	OIL	QUADRANT 11" LONG	1100	12	LONGITUDINAL	1" FROM CENTER	174000	120700	147000	15.0	38.0													
												1" FROM CENTER	168400	124800	136000	18.0	40.6	108900	70000	12.0	341 TO 364	66000		.29				
												3-1/8" FROM CENTER	168000	123000	138200	16.0	44.5												
												6" FROM CENTER	170800	130980	148400	14.9	41.2	109480	70890	12.0									
												6" FROM CENTER	169400	133700	140200	17.0	43.5	110000	69000	12.0									
												AVERAGE	170850	128950	142150	14.9	38.1	109450	69650	12.0									
											RADIAL	2" FROM CENTER	156700	130000		3.0	12.6													
												5" FROM CENTER	163400	122600	133700	7.0	7.5												
												8" FROM CENTER	165600	126600	136600	5.0	6.2												
												AVERAGE	161900	127090	134750	5.0	8.8												
									1200	4	LONGITUDINAL	1" FROM CENTER	125300	98900	103400	6.0	14.4												
												1" FROM CENTER	134700	102200	108100	14.0	27.5												
												3-1/8" FROM CENTER	130800	100300	102500	20.0	32.7	85980	54000	12.0	262 TO 289	55000	38000	.42	.30			
												3-1/8" FROM CENTER	133000	101200	105000	17.0	46.9												
												6" FROM CENTER	127900	100300	101800	21.0	33.5	87900	54000	12.1									
												6" FROM CENTER	130700	102100	104000	19.0	36.0	86000	56000	12.0									
												AVERAGE	130400	100950	104150	16.1	42.5	86250	55330	12.0									
											RADIAL	2" FROM CENTER	129800	101300	103200	9.5	14.8												
												2" FROM CENTER	128500	99400	101800	8.0	8.8												
												2" FROM CENTER	133700	104900	107700	8.0	8.9												
												AVERAGE	130650	101850	104050	8.5	10.9												
									1260	4	LONGITUDINAL	1" FROM CENTER	122300	89500	93300	23.0	56.2												
												1" FROM CENTER	125300	93200	94700	20.0	53.5												
												3-1/8" FROM CENTER	116900	87000	84500	23.0	58.9	80700	50000	12.0	252 TO 267	53000	34000	.44	.28			
												3-1/8" FROM CENTER	122000	90000	92500	19.0	56.0												
												6" FROM CENTER	119300	91100	92900	21.0	57.3	79700	52000	11.9									
												6" FROM CENTER	121800	94500	94700	22.0	56.2	79100	52000	12.0									
												AVERAGE	121800	90900	93600	21.1	55.8	79850	51300	12.0									
											RADIAL	2" FROM CENTER	119300	89500	91700	13.0	16.7												
												2" FROM CENTER	122600	90700	96000	18.0	13.4												
												2" FROM CENTER	118300	87000	89500	12.0	33.4												
												AVERAGE	120100	88950	92400	11.7	21.2												
									1300	4	LONGITUDINAL	1" FROM CENTER	116500	86300	89000	25.0	57.3												
												1" FROM CENTER	114500	83700	86200	21.0	47.8												
												3-1/8" FROM CENTER	110900	78300	82500	25.5	56.8	78700	44000	12.0	229 TO 236	50000	34000	.44	.30			
												3-1/8" FROM CENTER	113200	80200	81400	22.0	58.2												
												6" FROM CENTER	110800	78300	80900	23.0	58.8	79700	43200	12.0									
												6" FROM CENTER	112900	84500	85820	23.0	59.3	82000	45000	12.0									
												AVERAGE	113150	81500	85100	23.6	56.7	79450	44050	12.0									
											RADIAL	8" FROM CENTER	114000	82500	85000	16.0	21.5												
												8" FROM CENTER	106900	76300	79900	13.5	18.9												
												8" FROM CENTER	111900	77600	82000	15.0	20.2												
												AVERAGE	111600	78800	82150	14.8	20.3												
											LONGITUDINAL	1" FROM CENTER	113000	64200	68000	22.0	47.8												
												2" FROM CENTER	103700	57200	57700	25.5	54.3			23.4									
												5" FROM CENTER	106700	62500	63000	24.0	55.5	72600	26000	12.1	207 TO 212	45000	31000	.42	.29			
												3-1/8" FROM CENTER	102000	59400	60000	25.5	57.3			24.7									

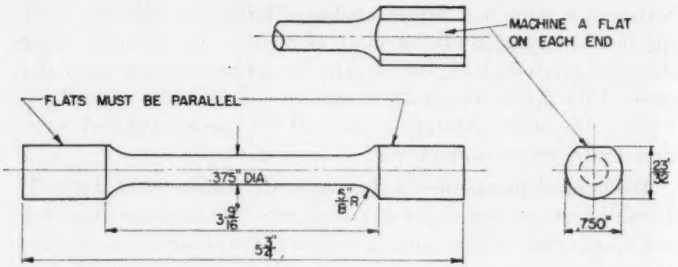
														AVERAGE	111600	76800	82150	14.6	30.3										
XL	9445	8" DIA X 36" LONG	1690	8		QUADRANT 11" LONG	1200	4	LONGITUDINAL	1" FROM CENTER	113000	84200	85000	22.0	47.8	73600	28000	12.1	23.4	207 TO 212	45000	31000	.42	.29
										2" FROM CENTER	103700	57200	57700	23.3	54.3												
										3" FROM CENTER	106700	62500	63000	24.0	55.5												
										3-1/2" FROM CENTER	102000	59400	60000	23.5	57.3				24.7								
										AVERAGE	106380	60800	61400	23.2	53.7				24.0								
						7" SQ. X 6-3/4"			TRANSVERSE	1" FROM OUTSIDE	120000	84800	86000	21.0	46.6				14.2								
										2" FROM OUTSIDE	111000	62200	62500	17.0	38.8				15.4								
										AVERAGE	115250	73550	73750	19.0	37.7				14.8								
						QUADRANT 11" LONG	1200		LONGITUDINAL	1" FROM CENTER	101500	56000	56500	27.0	54.5				21.9								
										2" FROM CENTER	100000	53500	57500	27.0	53.5				24.7								
										3" FROM CENTER	98000	58000	60000	30.0	58.0	71100	25400	12.0	25.7	197 TO 217	43000		.43				
										3-1/2" FROM CENTER	99500	61000	61500	29.0	58.6				26.1								
										AVERAGE	99750	57250	59300	28.2	56.3				24.3								
						7" SQ. X 6-3/4"			TRANSVERSE	1" FROM OUTSIDE	98000	38000	98400	21.6	31.7				14.0								
										2" FROM OUTSIDE	99500	57500	57700	24.0	41.6				14.9								
										AVERAGE	98250	58250	58250	22.7	36.6				14.9								
XI	9445	10" DIA X 36" LONG	1680	10		QUADRANT 11" LONG	1200	6	LONGITUDINAL	1" FROM CENTER	116500	54000	56700	16.0	27.5	63400	28000	12.0	10.7	235 TO 241	47000	32000	.40	.28
										2" FROM CENTER	117500	56200	58500	22.0	40.4				15.6								
										3" FROM CENTER	116500	54400	56300	22.0	39.4				14.2								
										3-1/2" FROM CENTER	115500	57500	58100	24.0	43.1				16.2								
										3-1/2" FROM CENTER	116500	58200	60000	22.0	46.5				16.7								
										AVERAGE	116500	56250	58300	21.7	39.4				14.3								
						7" SQ. X 6-3/4"			TRANSVERSE	1" FROM OUTSIDE	116000	57500	57700	20.0	32.1				11.9								
										2" FROM OUTSIDE	114000	53100	54100	17.9	23.7				12.9								
										AVERAGE	114500	55300	55900	18.7	27.9				12.3								
XK	9450	8" DIA X 36" LONG	1650	8		QUADRANT 11" LONG	1200	4	LONGITUDINAL	1" FROM CENTER	116200	48000	55700	18.5	35.7	85100	28800	12.0	16.7	237 TO 241	48000		.49				
										2" FROM CENTER	121500	60500	62500	19.5	43.6				18.2								
										3-1/2" FROM CENTER	122000	63000	63700	20.0	46.1				15.6								
										AVERAGE	119900	57200	60950	19.3	42.4				16.9								
						7" SQ. X 6-3/4"			TRANSVERSE	1" FROM OUTSIDE	120500	57500	59000	13.0	13.4												
										2" FROM OUTSIDE	118000	54200	57500	18.8	21.3												
										AVERAGE	119250	55850	58250	14.2	17.3												
						QUADRANT 11" LONG	1300		LONGITUDINAL	1" FROM CENTER	107500	50000	53700	22.0	39.4	79100	28800	12.0	16.4	215 TO 229	48500	34000	.42	.31
										2" FROM CENTER	114500	50500	53500	22.5	40.6				16.7								
										3" FROM CENTER	110700	55000	57500	25.0	49.2				16.7								
										3-1/2" FROM CENTER	113500	52500	56200	22.0	45.2				18.8								
										AVERAGE	106500	51950	55250	22.9	43.6				18.6								
						7" SQ. X 6-3/4"			TRANSVERSE	1" FROM OUTSIDE	104000	52500	53700	5.5	5.4												
										2" FROM OUTSIDE	115500	52500	53700	17.5	20.2												
										AVERAGE	109750	52700	53700														

National Emergency Steels analyzed failed to conform to th specified analysis in one or more elements. In instances where chemical analysis indicated a different grade of steel from that ordered the applicable grade designation is shown in parenthesis. Two of the steels, Materials VZ and VY, were modified intentionally by the manufacturer.

Mechanical properties of the steels are presented in Table II. Tension tests were made of standard .505-inch diameter threaded-end specimens. Static torsion tests were made of .75-inch diameter specimens. Charpy impact values were obtained from 10 mm. square keyhole notch specimen.

The fatigue tests in flexure were made with the rotating cantilever type of specimen. In this specimen the test length is conically tapered and so designed that the maximum stress is ¾-inch out from the inner fillet, and the stress varies only about 1.5 per cent over a length of 1.5 inches. All specimens were finished by a method of alternate longitudinal and transverse polishing. The surface finish by this method is sufficiently smooth to permit examination of the structure at a magnification of 100 diameters. The test length of the specimens was kept covered with a film of mineral oil throughout the test. The endurance machines were operated continuously at 1450 Rpm. except for the time required to remove and install specimens.

Endurance tests in alternating torsion were also made for some of the steels. Details of the alternating torsion machine are shown in Figure 1. One of the shafts is given an oscillating motion by means of a variable eccentric and opposing spring. This motion is transmitted through the test specimen and is opposed by a standard specimen of high elastic limit which acts as a torsional spring. The stationary end of the standard specimen is mounted in a grip which can be rotated and secured in any desired position. By this means, the standard and test specimens can be subjected to a pre-twist with the oscillating shaft on dead center. In this way the stress range can be varied. The stress is computed from the magnitude of twist obtained from the standard specimen. For the tests herein reported, the stress was maintained equal positive and negative from the neutral position. All tests were made at 1475 stress reversals per minute. Details of the alternating torsion endurance specimen are shown in Figure 2.



MODIFIED SPECIMEN FOR HIGH SPEED ALTERNATING TORSION MACHINE.

FIGURE 2

It will be noted that in general forgings 8 inches in diameter made of steels of the lower carbon ranges investigated can be normalized and drawn so as to conform to the requirements of Class AN, and those of the higher carbon ranges, of Class HG. For convenience of determining whether or not the steels in various conditions of heat treatment meet the physical requirements of Navy Department Specification 49S2 for Classes AN and HG, Table III was prepared. The designations used are either positive or negative; however, there are a number of marginal cases which can be ascertained by reference to Table II. It will be evident that for some of the steels specification requirements can be met by adjusting the tempering temperature.

The average endurance ratio (ratio of endurance limit to tensile strength) for the 20 steels tested in rotating flexure is .43, and the corresponding ratio in alternating torsion .29. While the average endurance ratio for the alternating torsion tests compares favorably with endurance ratios obtained for alloy steels such as 3½ per cent nickel steel, the average ratio for the rotating flexure tests is somewhat lower. In this respect, the NE steels tested behave more like carbon steels than alloy steels. The maximum endurance ratio was obtained for the 1¼-inch stock, Material VE, in the quenched and tempered conditions. For the normalized and tempered steels the best values for endurance ratio range from .44 to .47. In general, good endurance ratios were obtained for the 1300 degrees F temper.

Brinell readings were taken every half-inch along a center line

TABLE III

COMPLIANCE OF TENSILE PROPERTIES FOR NE STEELS WITH MINIMUM REQUIREMENTS OF NAVY DEPARTMENT SPECIFICATION 49 S2 FOR CLASSES AN AND MG

MAT'L LETTER	NE NO.	SIZE	NO SPECS 495-2 CL ASS	NORMALIZED												OIL QUENCHED					
				TEMPERING OR REHEATING TEMPERATURES DEG F												TEMPERING TEMPERATURES DEG F					
				1000	1100	1200	1250	1275	1300	1325	1350	1400	1450	1475	1500	1000	1100	1200	1260	1278	1300	1350
VK	8630	1" DIA	AN	YES		YES									YES	NO		YES			
			MG	YES		NO									NO	YES		YES			
WD		8" DIA	AN			YES			YES												
			MG			NO			NO												
VI		1" DIA	AN	YES		YES									YES	NO		YES			
			MG	YES		NO									NO	YES		YES			
XG		8" DIA	AN			YES															
			MG			NO															
VX	8739	1" DIA	AN		NO	YES			YES						YES			NO		NO	
			MG		YES	YES			YES						NO			YES		YES	
WE		8" DIA	AN			YES			YES												
			MG			NO			NO												
XF		8" DIA	AN			YES			YES												
			MG			NO			NO												
XO		1" DIA	AN		NO	NO			NO						YES		NO	NO		NO	
			MG		YES	YES			YES						YES		YES	YES		YES	
XP		1" DIA	AN		NO	NO			YES					YES			NO	NO		YES	
			MG		YES	YES			YES					YES			NO	YES		YES	
WG		8" DIA	AN	YES		YES									YES						
			MG	NO		NO									NO						
XM	8339	1" DIA	AN		NO	NO			YES						YES		NO	NO		NO	
			MG		YES	YES			YES						YES		NO	YES		YES	
WH		8" DIA	AN	YES		YES			YES						YES						
			MG	YES		NO			NO						NO						
VO		10" & 12"	AN	NORMALIZED & TEMPERED BY MFR DOES NOT MEET EITHER SPECIFICATION																	
			MG																		
VG	8744	1" DIA	AN	NO		NO									NO	NO		NO			
			MG	NO		NO									YES	NO		NO			
VF		8" DIA	AN	NO	NO			YES							YES						
			MG	YES	YES			YES							NO						
XN	8442	1" DIA	AN		NO	NO			YES						YES		NO	NO		YES	
			MG		YES	YES			YES						YES		NO	YES		YES	
WJ		8" DIA	AN	NO		YES			YES						YES						
			MG	YES		YES			NO						NO						
VE		1 1/2" DIA	AN												YES	NO		NO			
			MG												YES	NO		YES			
VJ	8749	1" DIA	AN	NO		NO									YES	NO		NO			
			MG	YES		YES									NO	NO		YES			
WY		9 3/8" DIA	AN			NO			NO												
			MG			YES			YES												
XJ		8" DIA	AN			YES			NO												
			MG			YES			YES												
WA		1" DIA	AN		NO	NO	NO			NO		NO									
			MG		NO	NO	NO			YES		NO									
VM		8" DIA	AN	NO		NO			NO												
			MG	NO		NO			NO												
WI		8" DIA	AN	NO		YES			YES						NO						
			MG	NO		YES			NO						NO						
XQ		1" DIA	AN	NO		NO			NO					YES		NO		NO		YES	
			MG	NO		YES			YES					NO		NO		NO		YES	
XU		1" DIA	AN		NO	NO			NO					YES			NO	NO		NO	
			MG		YES	YES			YES					YES			NO	NO		YES	
XD	8949	1" DIA	AN	NO	NO	NO				YES						NO	NO	NO	NO		YES
			MG	NO	YES	YES				YES						NO	NO	NO	YES		YES
XE		8" DIA	AN			YES			YES												
			MG			NO			NO												
VZ		1" DIA	AN		NO	NO			NO				YES					NO			
			MG		NO	NO			YES				NO					NO			
VY		8" DIA	AN			NO			YES					YES							
			MG			YES			NO					NO							
VN		1" DIA	AN		NO	NO			YES				YES					NO			
			MG		NO	YES			YES				YES					NO			
VD		13 1/2" DIA	AN						NO								NO	NO	NO		NO
			MG						NO								NO	NO	NO		NO
XL	9445	8" DIA	AN			NO			YES												
			MG			NO			NO												
XI		10" DIA	AN			NO															
			MG			NO															
XK	9450	8" DIA	AN			NO			NO												
			MG			NO			NO												

for the full cross section. Maximum and minimum hardness values are reported in Table II.

End-quench hardenability tests were made on all the steels, but inasmuch as they conform in general to the curves already published for these steels, it is not considered necessary to reproduce them here.

Likewise the results of hardness tests to determine the response to tempering are in close enough agreement with other published results to make their reproduction unnecessary.

Tests have also been conducted on steels which do not conform to the NE analyses but contain less than 2.75 per cent minimum nickel. One such test was made of a 14-inch diameter forging of the following composition: Carbon—0.45 per cent, Manganese—0.75 per cent, Silicon—0.20 per cent, Nickel—1.38 per cent, Vanadium—0.07 per cent. While the results of this test are not complete, those which have been obtained to date follow:

	Tensile Strength P. S. I.	Proof Stress P. S. I.	Elongation in 2 ins. %	Reduction of Area %	Brinell Hardness
Average.....	89,300	54,800	31.8	58.4	181
Minimum....	87,200	48,000	30.0	45.1	179

The endurance limit in alternating torsion was 26,500 P. S. I., which corresponds to a ratio of .30 of the average tensile strength. This is equal to that obtained for Class AN 2.75 per cent minimum nickel steel. As far as the tests have progressed it appears that the strength characteristics of this low nickel-vanadium steel compare favorably with the higher nickel steel.

Since some of the NE Steels as well as SAE 4140 steel had been proposed for use as high temperature bolting materials, it was desirable to investigate their high temperature properties at the same time that the general study was being made. The chemical analyses of the steels selected are given in Table IV with the results of various tests in Tables V, VI, VII, and VIII.

The static tension test results show that neither the NE steels nor SAE 4140 steel meet the requirements of Navy Department Specification 43B14C for high temperature bolting material with regard to tensile strength when normalized and tempered at

TABLE IV

CHEMICAL COMPOSITION

EES Mat'l desig.	NE Steel desig.	Bar desig.	Chemical Composition - % C	S	P	Mn	Si	Cr	Ni	Mo	W	V
WZ	SAE 4140	A	.46	.029	.020	.92	.21	1.06		.20		
		B	.45	.025	.022	.93	.21	1.03		.22		
		C	.45	.027	.022	.92	.21	1.05		.22		
VX	NE 8739	1	.36	.01	.018	.83	.30	.50	.42	.22		
VJ	NE 8749	1	.52	.01	.01	.85	.26	.56	.35	.20		
WA	NE 8749	2	.40	.049	.02	.90	.22	.50	.60	.46		
VZ	NE 8949	1	.45	.022	.03	.84	.20	.58	.54	.34		
XD	NE 8949	1	.44	.012	.022	1.10	.25	.54	.46	.33		
QG		---	.41	.013	.014	.44	.26	.93		.48		.22
QWA	Cr-MoV	---	.38	.010	.012	.64	.19	1.02		.48		.24
QV		---	.39	.050	.021	.76	.25	.38		.59	1.08	
QVA	Cr-Mo-W	---	.38	.020	.023	.76	.24	.63		.60	1.19	
Navy Dept. Specification 43B-14C (1)			Max. .50	Max. .045	Max. .04			Max. 2.0		(2) Min. .45	(2) Min. .75	

(1) Steels containing additional elements such as Cr, W, Ni or V may be used
(2) Tungsten or molybdenum

1200 degrees F., but do meet them in regard to ductility. These steels will meet the requirements of this specification in regard to tensile strength, proof stress, and ductility when oil quenched from 1550 degrees F. and tempered at 1200 degrees F.

Test results obtained thus far at 950 degrees F. temperature show that the stress-rupture strength values for the NE steels and SAE 4140 in the normalized and tempered condition are superior to those obtained for the oil quenched and tempered condition. The stress-rupture values obtained for the alternate bolting materials increase with increase in molybdenum content, but are inferior to the standard bolt steel, material QWA.

Similarly, creep test results obtained for SAE 4140 in the normalized and tempered condition are superior to those obtained for the material in the oil quenched and tempered condition.

Although a number of creep tests are still in progress, the results indicate that SAE 4140 and the NE steels investigated do not have the high temperature strength properties required to meet Navy Department Specification 43B14C. Creep resistance is enhanced particularly by increasing the molybdenum content. For this reason, material WA has indicated the maximum creep

TABLE V

RESULTS OF TENSION TESTS AT ROOM TEMPERATURE FOR SAE 4140 STEEL-WZ

TEMPERING TEMP. DEG. F.	TENSILE STRENGTH	YIELD STRENGTH (.1% SET)	PROOF STRESS	ELASTIC LIMIT	PROP. LIMIT	ELONG. IN 2 IN	RED. OF AREA
	LB. PER SQ. INCH					PER CENT	
NORMALIZED AT 1650° F. AND TEMPERED AS INDICATED							
1000	140000	106500	98000	90000	87500	20.0	53.3
	143000	105300	95000	87000	85000	19.0	51.0
AVERAGE	141500	105900	96500	88500	86250	19.5	52.2
1100	139300	105800	97000	85000	85000	19.0	53.6
	139000	103000	95000	86000	82500	20.0	51.0
AVERAGE	139150	104400	96000	85500	83750	19.5	52.3
1200	116000		86200	85300		21.0	59.8
	117000		85600	85000		21.0	60.1
AVERAGE	116500		85900	85150		21.0	60.0
1300	107000	75000	72000	70000	65000	22.0	63.6
	109000	76000	74500	71000	70000	20.0	60.0
AVERAGE	108000	75500	73250	70500	67500	21.0	61.8
1400	99000	57500	56000	55000	52500	28.5	62.0
	96000	55000	54000	52500	42500	25.0	65.0
AVERAGE	97500	56250	55000	53750	47500	26.8	63.5
1550	109250	57500	56000	55000	50000	22.0	48.5
	108500	60000	58500	55000	52500	22.0	47.8
AVERAGE	108900	58750	57250	55000	51250	22.0	48.2
NORMALIZED AT 1600°F, OIL QUENCHED FROM 1550 °F, TEMPERED AS INDICATED							
1000	194000	180000	175000	168000	165000	14.0	49.1
	194000	178000	171000	163000	160000	13.0	48.0
AVERAGE	194000	179000	173000	165500	162500	13.5	48.6
1100	178200	164000	159000	152000	145000	17.0	47.2
	177000	165500	158000	151000	145000	15.5	50.3
AVERAGE	177600	164750	158500	151500	145000	16.3	48.8
1200	135500		126500	125000		21.0	60.6
	137000		128100	126800		21.0	60.7
AVERAGE	136250		127300	125900		21.0	60.7
1250	128700		119500	118700	117500	24.0	59.8
1300	115000		101000	100500	100000	20.5	63.5
	122000		114000	113000	112500	22.0	60.3
	111300	107000	107000	105000	104000	25.0	64.4
	124000	112500	109500	107000	90000	27.0	63.8
	124000	110500	109000	105000	100000	30.0	62.3
AVERAGE	118600	110000	108100	106100	101300	24.9	62.9
1400	95000	77400	77000	75000	70000	29.0	69.0
	96750	74800	73000	70000	70000	27.0	66.1
AVERAGE	95900	75850	75000	72500	70000	28.0	67.6
1550	110000	57100	56000	51000	40000	21.0	47.8
	110000	59800	58000	47500	45000	22.5	48.9
AVERAGE	110000	58450	57000	49250	42500	21.8	48.4

TABLE VI -

RESULTS OF TENSION TESTS AT ROOM AND ELEVATED TEMPERATURES - BOLT STEELS

STEEL	EES DESIGNATION	HEAT TREATMENT	TESTING TEMP DEG FAHR	TENSILE STRENGTH	YIELD STRENGTH (1% SET)	PROOF STRESS	PROP LIMIT	ELONG IN 2 IN	RED OF AREA	MODULUS OF ELASTICITY (NOMINAL)
				LB PER SQ INCH				PER CENT		LB PER SQ IN
SAE 4140 ASTM B-7 1" DIA STOCK	WZ	N 1650° F T 1200° F	ROOM	115500	86500	86000	75000	21.0	59.3	31000000
			850	90000	62500		43800	29.0	77.6	
			950	77000	60000		36300	34.5	83.2	
			1000	73000	50000		33000	35.5	86.2	19400000
			1100	57000	34500		23500	38.5	86.3	
			1200	34000	15000		6000	47.0	88.9	8800000
		N 1600°F O 1550°F T 1250°F	ROOM	128700	119500	119500	117500	24.0	59.8	29100000
			850	101500	70500			27.0	79.6	
			950	85500	51500		32500	24.5	86.4	
			1000	81500	42500		18500	27.0	88.4	17000000
		N 1600°F O 1550°F T 1300°F	ROOM	118600	110000	108100	101300	24.9	62.9	30200000
			850	83000	58700		33800	25.0	75.9	22700000
			1000	71000	41000		17500	31.0	88.5	18900000
NE 8739 1" DIA STOCK	VX	N 1600° F T 1200° F	ROOM	105000	84700	81500	75000	27.0	58.5	30500000
			750	90500	58800		48700	30.5	72.2	24200000
			850	81000	55000		40000	32.5	73.5	
			950	72000	50000		31000	34.0	79.2	21900000
			1000	65500	43200		24000	37.5	83.2	17600000
			1100	53000	30000		18000	38.0	84.7	15100000
			1200	34100	15200		10000	61.0	81.6	
		N 1600° F O 1550° F T 1200° F	ROOM	128400	118500	117000		23.3	61.3	
NE 8749 1" DIA STOCK	VJ	N 1600° F T 1200° F	ROOM	112700	84500	83900	81400	24.2	57.2	29500000
			850	84500	55600		37000	30.0	75.3	24300000
			950	73800	46900		28000	32.7	78.1	
			1000	68200	47000		26000	39.8	81.8	19900000
			1100	52500	31500		14500	49.0	85.0	15000000
			1200	35500	15000*			37.0	82.0	
		N 1600° F O 1550° F T 1200° F	ROOM	130600	120000	119200		24.0	63.0	
NE 8749 1" DIA STOCK	WA	N 1600° F T 1200° F	ROOM	151500	122800	118400	110300	18.5	55.3	29100000
			850	120700	92400		62000	18.0	59.0	25100000
			950	107000	86300		48700	20.0	63.5	24100000
			1000	87000	65000		27500	25.0	77.4	19700000
NE 8949 1" DIA STOCK	VZ	N 1600° F T 1200° F	ROOM	123300	91700	89700	80500	20.5	54.8	30100000
			750	112000	82500		63700	22.0	60.6	23700000
			850	98000	74400		58700	22.5	64.2	
			950	92500	70000		48700	22.5	65.9	21500000
			1000	86000	65000		42500	23.0	69.2	18300000
			1100	61000	45000		25000	29.5	77.2	17800000
			1200	47000	25500		15000	36.0	86.1	11800000
		N 1600° F Q 1550° F T 1200° F	ROOM	145200	13700	129700		20.5	58.6	
NE 8949 1" DIA STOCK	XD	N 1600° F T 1200° F	ROOM	119000	89500	89200	87400	22.5	58.5	29000000
		N 1600° F O 1550° F T 1275° F	ROOM	128700	122500	122000	118000	21.0	61.8	30000000
			850	91000	65700		33800	27.0	74.6	25000000
			950	84000	60100		31200	25.0	85.0	22300000
Cr-Mo-V ASTM B-14 1" DIA STOCK	QG	AS REC'D	ROOM	144300	120000	117000	96700	19.5	54.2	30800000
	OWA	N 1725° F T 1200° F	ROOM	144900	121000	120000	100000	17.8	55.5	30300000
			850	112000	86200		62500	21.3	66.3	
			950	98700	79200		45000	20.7	69.9	23700000
			1000	94500	73500			25.0	71.4	
			1100	80500	49400			26.0	77.3	
			1200	59000				25.5	69.8	
Cr-Mo-W ASTM B-13 7/8"DIA STOCK	QVA	N 1725° F T 1200° F	ROOM	129100	95000	91700	67500	21.8	57.9	30600000
			850	113500	84400		56000	20.5	60.8	
			950	107500	80000		35000	16.8	58.1	24900000
			1000	89000	70000			20.5	68.3	20400000
			1100	80500	58700			16.5	67.4	
			1200	40000	21000			28.5	83.6	
Cr-Mo-W 1" DIA STOCK	QV	AS RECEIVED	ROOM	121700	120000	115000	105000	19.0	57.0	30800000
			950	77500	60000		37500	28.0	75.8	21200000
NAVY DEPARTMENT - SPECIFICATION 43 B 14c			ROOM	MIN 120000		MIN 100000		MIN 18.0	MIN 50.0	

N - NORMALIZED O - OIL QUENCHED T - TEMPERED * - APPROXIMATE

TABLE VII

RESULTS OF STRESS RUPTURE TESTS

STEEL	EES DESIGNATION	HEAT TREATMENT	TEMP. DEG. FAHR.	STRESS PSI	TIME TO FRACTURE, HOURS	ELONG. IN 2 IN.	RED. OF AREA	STRESS TO FRACTURE IN—HOURS, PSI	
						PER CENT		100	1000
SAE 4140 ASTM A-7 1" DIA. STOCK	WZ	N. 1650° F. T. 1200° F.	850	55000	1845	31.5	75.9	64000	56000
			950	50000	48.0	45.5	82.7	47000	40000 *
				45000	219	37.4	77.0		
		N. 1600° F. O. 1550° F. T. 1360° F.	950	50000	50.4	44.5	86.1	46000	35000
				42000	204.7	48.5	86.1		
				38000	535	60.5	86.5		
		N. 1600° F. O. 1550° F. T. 1300° F.	950	50000	20.2	35.5	83.9	44000	38000
				40000	530.3	53.0	86.3		
NE 8749 1" DIA. STOCK	VJ	N. 1600° F. T. 1200° F.	950	50000	64.9	43.5	69.9	45000	42000
				45000	106	32.4	64.3		
				40000	REMOVED AFTER 2700 HRS.				
NE 8749 1" DIA. STOCK	WA	N. 1600° F. T. 1200° F.	950	75000	14.7	21.0	71.6	66000	53000
				65000	460	22.0	48.6		
				60000	549	20.5	50.5		
				55000	869	19.5	48.0		
NE 8949 1" DIA. STOCK	VZ	N. 1600° F. T. 1200° F.	950	60000	901	12.0	23.6	61000	52000
				55000	719	12.2	19.3		
				45000 — 50000	1440 — 356	8.5	10.3		
				45000	3364	6.5	9.5		
		N. 1600° F. O. 1550° F. T. 1300° F.	950	55000	9.6	22.0	77.4	48000	42000
				45000	320	25.0	73.2		
NE 8949 1" DIA. STOCK	XD	N. 1600° F. T. 1280° F.	950	55000	29.6	31.5	78.0	50000	45000
				45000	1218	10.0	14.0		
			1000	45000	102.9	34.0	48.6	45000	
		N. 1600° F. O. 1550° F. T. 1275° F.	950	58000	37.8		79.8	48000	* 37000
				45000	147	37.0	71.3		
Cr-Mo-V ASTM A-14 1" DIA. STOCK	QWA	N. 1725° F. T. 1200° F.	950	75000	108	19.0	62.4	75000	61000
				70000	454	8.0	23.0		
				65000	234	17.3	31.9		
				60000 — 52000	2150 — 429	3.0	3.9		
			1000	55000	241	11.0	21.3	68000	43000
				45000	856	9.5	10.3		
				35800 — 40000 — 45000	216 — 48 — 32	3.5	2.2		
Cr-Mo-W ASTM A-13 1" DIA. STOCK	QVA	N. 1725° F. T. 1200° F.	950	55000	501	1.0	1.8	62000	50000
				48000	1674	1.8	1.8		
			1000	40000	1173	1.0	0.4		

N. - NORMALIZED O. - OIL QUENCHED T. - TEMPERED * - EXTRAPOLATED

TABLE VIII

RESULTS OF CREEP TESTS

STEEL	EES DESIGNATION	HEAT TREATMENT	TEMP DEG. FAHR.	STRESS PSI	TOTAL TIME HOURS	TOTAL CREEP IN./IN.	CREEP RATE TIME HOURS	CREEP RATE RATE % / HR.	STRESS FOR .1% CREEP IN 10000 HOURS, PSI
SAE 4140 ASTM B-7 1" DIA. STOCK	WZ	N. 1650° F. T. 1200° F.	850	30000	IN PROGRESS				
			950	20000	2188	.00572	250-2188	.000129	
				15000	IN PROGRESS				
			1000	15000	650	.01048	0-650	.00161	
		N. 1600° F. Q. 1550° F. T. 1250° F.	950	15000	336	.00708	50-336	.09907	
		N. 1600° F. Q. 1550° F. T. 1300° F.	950	15000	480	.00815	50-480	.001298	
NE 8739 1" DIA. STOCK	VX	N. 1600° F. T. 1200° F.	850	30000	2708	.00159	250-2708	.000038	
				25000	IN PROGRESS				
NE 8749 1" DIA. STOCK	VJ	N. 1600° F. T. 1200° F.	850	30000	2088	.00291	750-2088	.000067	
			950	25000	288	.01265	0-288	.00439	
			1000	15000	1320	.00872	250-1320	.000556	
				20000	288	.00791	0-288	.002746	
NE 8749 1" DIA. STOCK	WA	N. 1600° F. T. 1200° F.	950	25000	IN PROGRESS				
NE 8949 1" DIA. STOCK	VZ	N. 1600° F. T. 1200° F.	950	20000	IN PROGRESS				
			1000	18000	1763	.00410	250-1763	.000262	
NE 8949 1" DIA. STOCK	XD	N. 1600° F. T. 1200° F.	950	20000	IN PROGRESS				
Cr-Mo-V ASTM B-14	QG 7/8" DIA. STOCK	AS RECEIVED	850	40000	1700	.00026	250-1700	.0000069	45000
				45000	3334	.00076	250-3334	.0000084	
			950	25000	5160	.00075	250-5160	.0000063	27000
				30000	5200	.00144	250-5200	.000017	
				35000	5210	.00236	250-5210	.0000314	
			1050	10000	4470	.00181	250-4470	.0000317	
				15000	3216	.00592	250-3216	.000157	7500
				25000	480	.00757	0-480	.001577	
	QWA 7/8" DIA. STOCK	N. 1725° F. T. 1200° F.	850	30000	3000	.00075	1000-3000	.000014	
			950	25000	3456	.00123	250-3456	.000019	24500 APPR.
Cr-Mo-W ASTM B-13	QV 1" DIA. STOCK	AS RECEIVED	950	10000	3072	.00324	250-3072	.000084	
				14000	2712	.00560	250-2712	.000170	7000 APPROX.
				20000	288	.00531	0-288	.001844	
				25000	190	.00731	0-190	.003847	
	QVA 7/8" DIA. STOCK	N. 1725° F. T. 1200° F.	950	20000	3288	.00060	250-3288	.000008	16800 APPROX.

N - NORMALIZED
O - OIL QUENCHED
T - TEMPERED

resistance of the NE steels tested. Material WA cannot be regarded as an NE steel, but meets the chemical requirements of Navy Department Specification 43B14C. Likewise, the molybdenum contents for materials VZ and XD are slightly above the maximum of the range for NE steels, namely .20 to .30 per cent.

Conclusions.

National Emergency Steels of the 8000 series herein described offer much promise as alternate materials for forgings, classes AN and HG, of Navy Department Specification 49S2. Of the seven steels of this series, one is of the semi-thorough hardening grade, (NE 8630) and the others of the thorough-hardening grade. All have shown to good advantage when properly heat treated. Of the less hardenable steels, NE 8739 is a good representative material. Of the steels of greater hardenability, NE 8749 has shown good properties.

These steels are well adapted for heat treatment by normalizing and tempering. This procedure is desirable for large sections and the tempering temperature is not critical. The tempering temperature is more critical after oil quenching. In general, desirable properties can be obtained by normalizing and tempering at 1200 degrees F.–1300 degrees F.

Steels NE 9445 and 9450 are less satisfactory. These steels failed to meet either AN or HG requirements when normalized and tempered at 1200 degrees F. However, one of the 9445 steels meets AN requirements when normalized and tempered at 1300 degrees F. Steel NE 9450 failed to meet either AN or HG requirements when normalized and tempered at 1200 degrees and 1300 degrees F.

None of the samples showed any excessive amounts of nonmetallic inclusions. Likewise, no flakes or cracks were encountered in the forgings.

The average endurance ratio (ratio of endurance limit to tensile strength) for the 20 steels tested in rotating flexure is .43. This is somewhat lower than would be expected for alloy steels, and in this respect the steels are intermediate between carbon and alloy steels. In regard to similar ratios for endurance in alternating torsion, these are not appreciably different from those for alloy steels.

The NE steels of this series are inferior to available high temperature bolting materials. In this type of material the high temperature strength depends upon molybdenum or tungsten. The NE steels contain no tungsten and the molybdenum content is not sufficiently high for the necessary strength at elevated temperatures.

At the present stage of investigation, it appears that some of the NE steels, as well as other steels containing smaller amounts of alloys than were customarily used under peace time conditions, are capable of developing properties which are satisfactory for use in some applications in the Naval service.

NOTES.

INDEX TO NOTES.

THE COMBUSTION GAS TURBINE.
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How Can We Develop Inventors?
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THE COMBUSTION GAS TURBINE.

Month by month, the volume of literature on the combustion gas turbine in the technical press is increasing by leaps and bounds. This particular article by F. K. Fischer and C. A. Meyer, both engineers employed by the Westinghouse Electric and Manufacturing Company, gives a broad picture of the situation today. It is reprinted from the Marine Journal, May, 1944. A companion article is to appear in that publication in another issue.

There are many future uses of the gas turbine because it potentially promises higher efficiency at very high temperatures than most engines and prime movers used today. Future applications range from electric power generation to power-plants for propelling airplanes, trains and ships.

Advantages of the gas turbine cycle as compared to the conventional steam system include: (1) no boiler is used; (2) water is not required for the simple open-cycle system; (3) promises greater efficiency improvement at high temperature and (4) high hp-per-lb. output for short life applications. Present disadvantages include: over-optimism; fuel limited to high grade oils instead of low grade oil and coal; little field experience and need of time to complete technical developments in metallurgy and component parts of the gas turbine system. To a large degree, the future application of the gas turbine depends upon developments in the field of metallurgy, aerodynamics, combustion and heat exchange. Present knowledge in these fields permits building and operating simple gas turbines for certain purposes. Experience with some of the simple forms of gas turbine plants has been successful and encouraging. In the post-war period, industry will benefit from the developments now being engineered for national defense. These developments will accelerate the application of gas turbine plants to new and larger fields.

Gas Turbine Is Simple.

The fundamental directness of the gas turbine power cycle, in which all the hot gases of combustion are led straight to the turbine, has intrigued engineers for years. Versatile Leonardo da Vinci devised a crude version. In 1791, John Barber, an Englishman, took out the first patent on a turbine operated by gases. Since that time there has been an almost continuous stream of developments.

Progress toward a practical gas turbine power unit has been delayed because the thermal efficiency required to make it competitive with the highly developed steam cycle required: (1) operation above 1000 degrees F.; (2) a highly efficient compressor; and (3) a highly efficient turbine. Two seemingly unrelated industries have recently made important contributions to help solve these problems. Metallurgists, in developing materials for superchargers have produced alloys that are expected to withstand at least 1200 degrees F. continuous service at the low operating pressures encountered in gas turbine work. Aviation and wind tunnel research on airplane propellers and wings have contributed fundamental aerodynamic data on which high efficiency compressor designs are based. The research in these two industries plus the accumulated steam experience of many years has made possible the necessary high-efficiency turbine and compressor elements and solved many of the mechanical problems involved in gas turbines.

The open cycle combustion gas turbine eliminates a big and expensive step in providing power to drive generators or machinery. In its power cycle, all the hot gases of combustion go straight to the turbine, thus eliminating the steam boiler. By contrast, in the highly developed steam cycle, the fuel energy must be converted to steam, at a comparatively low temperature, before expending its energy in the turbine. In the combustion gas turbine system of power generation there are two basic cycles—the open cycle for moderate capacities and the closed cycle for very large units.

The gas turbine employs the simplest power cycle known, consisting of three major elements: a compressor, a combustor and a gas turbine. A gen-

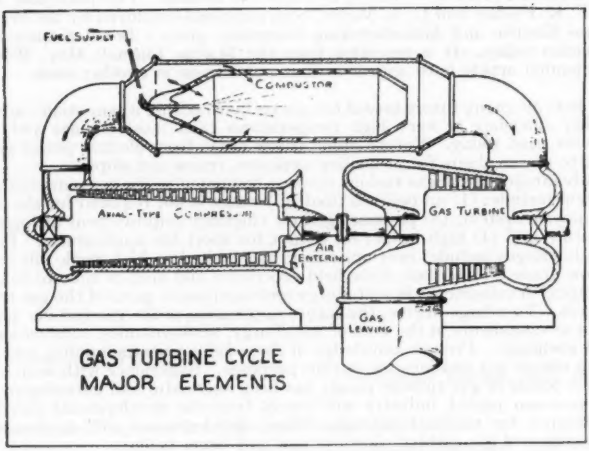


FIGURE 1.

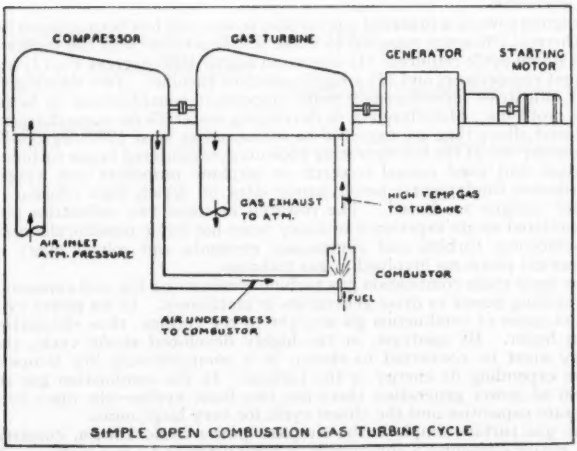


FIGURE 2.

eral idea of what the elements are like are shown as longitudinal sections in true relative size in Figure 1. The gas turbine resembles the straight reaction, non-condensing steam turbine. Gas turbine blades look more like air foil sections than reaction steam turbine blades due to the small pressure drop and large gas volume involved. The axial-flow compressor also resembles a straight reaction turbine, with the gas to be compressed passing axially through the compressor. The action of the blades in the axial-flow compressor is the reverse of the action of expansion in a reaction turbine. This physically small compressor handles the large volume of gas efficiently. The combustor is the burner in which the chemical energy of the fuel oil is converted into heat energy, by burning the fuel with sufficient excess air to obtain the desired temperature. The hot product of combustion from the combustor is the gas, which, expanding to a lower pressure and temperature in the gas turbine, converts some of its heat energy into mechanical energy at the turbine shaft. The combustor is relatively small since its rate of heat release is many times that of the conventional steam boiler which, in addition to burning fuel, must transfer the heat through tube walls to generate steam.

EFFECT OF DIFFERENT OPEN CYCLE ARRANGEMENTS ON EACH MAJOR ELEMENT.

Item	Simple Cycle	Cycle with Regeneration	Intercool and Regenerate	Reheat and Regenerate	Intercool Reheat Regenerate
Figure No.	2	4	5	6	7
Input in Fuel	4.95	3.75	3.43	3.55	3.11
Turbine Rating	3.95	2.95	2.80	2.88	2.55
Compressor Power	2.95	1.95	1.80	1.88	1.55
Useful Output	1.00	1.00	1.00	1.00	1.00
Per cent Eff. at 1200°F	20.2	26.6	29.2	28.1	32.2
Gas Temp. in °F. at					
Turbine Inlet	1200	1200	1200	1200	1200
Leaving Reheater	...	...	...	1200	1200
Turbine Exhaust	635	790	695	920	865
Leaving Regenerator	...	455	350	560	520
Air Temp. in °F. at					
Compressor Inlet	70	70	70	70	70
Leaving Intercooler	...	...	70	...	...
Leaving Compressor	490	340	230	440	405
Entering Combustor	490	680	575	800	750
Press. lbs./sq. in. abs at					
Compressor Inlet	14.7	14.7	14 7	14.7	14.7
Compressor Discharge	88.2	51.5	73.5	73.5	102.9
Turbine Inlet	88.2	50.2	71.7	71.7	100.4
Turbine Exhaust	14.7	15.1	15.1	15.1	15.1

HOW CYCLE WORKS.

The combustion gas turbine cycle in its basic form comprises three major elements previously mentioned: compressor, combustor and gas turbine; plus a generator or shaft for transmitting the useful power output, and a means of starting. This simple arrangement is called the open cycle system. To start a combustion gas turbine some external means, such as a motor, is required. This is necessary as the air for combustion is supplied to the combustor by the compressor. When the unit is in operation the energy to drive the compressor comes from the expansion of the products of combustion in the gas turbine.

In the simple cycle operating at 1200 degrees F., the products of combustion (gas) contain some 600 per cent excess air. This gas is expanded in the turbine and exhausted to the atmosphere. No intermediate fluid is used as in the steam cycle, where fuel is burned and releases chemical energy as hot products of combustion which pass through a steam boiler to generate steam for the steam turbine. The steam generator, or boiler, is the biggest single element in the steam power plant. In the gas cycle a compressor and combustor replace the boiler, resulting in a much smaller and more compact power plant. Because the simple open combustion gas cycle does not require cooling water, no steam condenser such as used in the condensing steam cycle is needed and plants may be located without regard to a suitable source of cooling water.

Although the first patent was taken out over 150 years ago, early inventors were unsuccessful in getting units efficient enough to drive their own compressor. The cycle was perfectly sound, but, these early inventors lacked materials to withstand the necessary high temperature plus turbines and compressors of suitable efficiency.

High-Temperature Metals Developed.

Today the story is changing. Materials are capable of operating at high temperatures. The maximum temperature employed is largely a question of the useful life. For a life of relatively few hours, such as might be satisfactory for some military needs, temperatures of the order of 2000 degrees F. are allowable. Heavy duty or long life applications are limited to much lower temperatures. The tremendous advances in metallurgy for the war effort will undoubtedly produce materials capable of operation at temperatures which we would not have attained for many years under a normal peace time development.

The combustion gas turbine required tremendous volumes of gas. The axial flow compressor, small in physical size, efficiently handles the large volumes required. Until recently the axial flow compressor was largely an unknown quantity. Today, the advancement in knowledge of proper blade shapes and research in the development of efficient air foil sections have given the designer the necessary tools to design and build highly efficient axial flow compressors.

Turbine and Compressor Efficiency Is Big Factor.

In the gas turbine cycle, the useful power output depends upon obtaining relatively high turbine and compressor efficiency. The useful net output is the difference of two fairly large quantities: the total turbine output and the work consumed by the compressor in compressing the air. With the system of Figure 2 operating at a top temperature of 1200 degrees F.; for each useful unit of power output, the turbine develops 3.95 units of which 2.95 is needed to drive the compressor. A reduction of 1 per cent in the efficiency of each of these elements reduces the useful output by 7 per cent. The useful output decreases sharply as the top temperature is lowered.

In Figure 3 the simple gas cycle thermal efficiency is plotted as a function of the pressure ratio for several turbine inlet temperatures of the combustion gas. Pressure ratio is the compressor discharge pressure divided by the compressor inlet pressure, which is 14.7 pounds per square inch absolute in the open cycle. The curves are based on these efficiencies: Turbine 85 per cent; compressor 84 per cent; combustor 100 per cent; and air inlet temperature of 70 degrees F. The 750 degrees F. curve is plotted for two different compressor efficiencies showing the effect of change in efficiency of one of the elements. A reduction in compressor efficiency of 9 per cent cuts the cycle efficiency almost in half. The relatively low pressure ratios at which maximum efficiency is obtained, around 6 for 1200 degrees F., means that relatively low turbine inlet pressures exist in the open cycle, under 100 pounds per square inch absolute. Low pressure results in large gas volume and relatively large turbine blade dimensions.

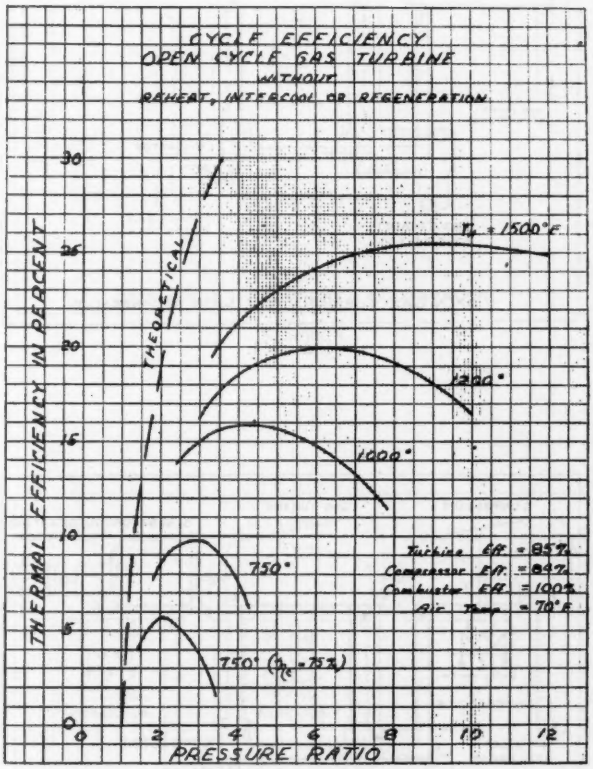

CYCLE EFFICIENCY
OPEN CYCLE GAS TURBINE
WITHOUT
REHEAT, INTERCOOL OR REGENERATION
THERMAL EFFICIENCY IN PERCENT
PRESSURE RATIO
THEORETICAL
T_4 = 1500°F
1250°
1000°
750°
750° (η_{tc} = 75%)
Turbine Eff. = 85%
Compressor Eff. = 80%
Combustor Eff. = 100%
Air Temp = 70°F
0
5
10
15
20
25
30
2
4
6
8
10
12

FIGURE 3.

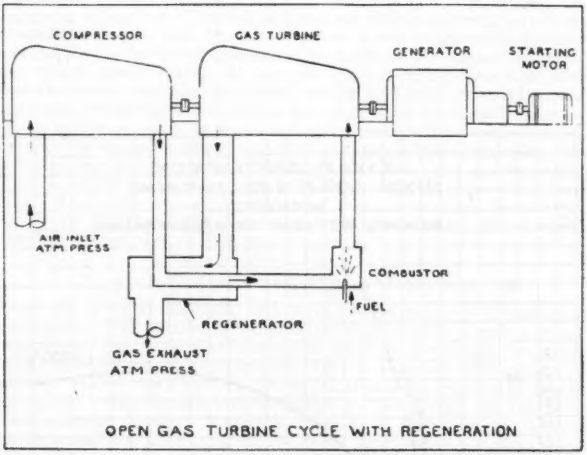


FIGURE 4.

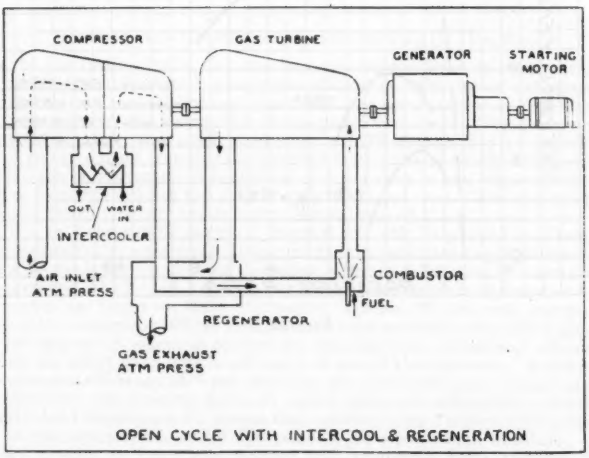


FIGURE 5.

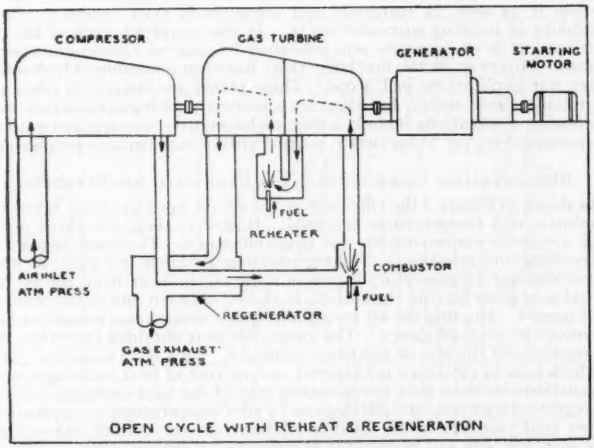


FIGURE 6.

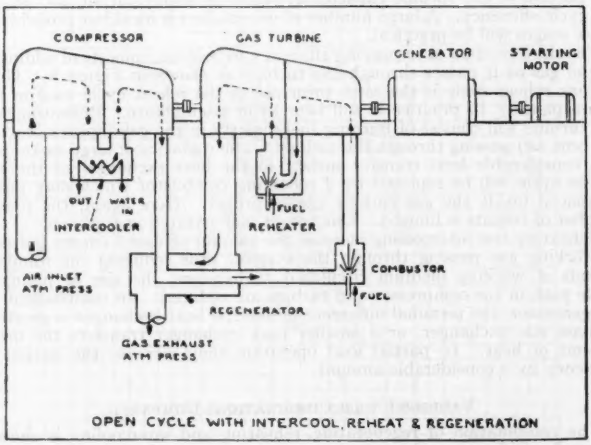


FIGURE 7.

Engineers have been experimenting with this combustion gas cycle for many years. In Europe, because of economic conditions engineers attempted to develop it as soon as materials and compressors even remotely offered a possibility of building successful units. In this country, much of the recent development is of necessity not published because of Government security reasons. However, in the literature there has been a considerable description of pre-war applications in Europe. These plants are inferior in efficiency to what can be built today, but they are apparently giving satisfactory results. Interesting applications include a turbine locomotive, electric generating set, compressor drive for Velox boiler, refinery drives and airplane propulsion.

Regenerating, Intercooling, and Reheating Aid Efficiency.

As shown in Figure 3 the efficiency of the simple open gas cycle is low unless extremely high temperatures are used. However, there are three practical ways of greatly improving the gas cycle efficiency. They are: regenerating, intercooling and reheating. The regenerating gas cycle is a cycle in which a heat exchanger (regenerator) transfers some of the heat from the relatively hot exhaust gases leaving the turbine to the air before it enters the combustor. See Figure 4. Heating the air by exhaust gases reduces fuel consumption and improves the cycle efficiency. The amount of heat obtained from the exhaust gas depends on the size of the heat exchanger. This is an economic problem in which gain in efficiency is balanced against cost of heat exchanger surface. Calculations indicate that the economic size of the heat exchanger will limit the regenerating cycle, at 1200 degrees F. inlet temperature, to approximately 75 per cent recovery of the heat available from the turbine exhaust gases. This economic size will be about 0.30 cubic foot of heat exchanger volume per Kw. of capacity.

Efficiency is further improved when intercooling is added to regeneration as shown in Figure 5. As the name implies, intercooling removes the heat of compression from the air passing through the compressor. Water, circulating through the intercooler cools the air, and is a necessary part of the cycle. By intercooling, the compressor work is reduced because the colder air has smaller volume. Other conditions remaining the same, one stage of intercooling will reduce the compressor work by some 15 per cent. This increases the portion of the turbine capacity available as useful output and improves the cycle efficiency. A large number of intercoolers is ideal but probably only a few stages will be practical.

The third method of improving efficiency, reheating, consists of adding heat to the gas as it passes through the turbine as shown in Figure 6. The gas turbine reheat cycle is the same principle as the reheat cycle used in many steam plants. In practice it will bear little resemblance. Reheating in the gas turbine will consist of burning fuel directly in the gas (approximately 85 per cent air) passing through the turbine. The quantity of large steam piping and considerable heat transfer surface in the heat exchanger of the steam reheat cycle will be replaced by a reheating combustor which may possibly be placed inside the gas turbine casing proper. Here again, the practical number of reheats is limited. One reheat is illustrated in Figure 6.

Reheating and intercooling increase the amount of useful energy per pound of working gas passing through the system, thus reducing the number of pounds of working medium circulated. Therefore, the size of piping and blade path in the compressor and turbine are reduced. In combination with a regenerator, the terminal difference across the heat exchanger is greater for a given size exchanger, or a smaller heat exchanger transfers the original amount of heat. In partial load operation they increase the partial load efficiency by a considerable amount.

Various Cycle Combinations Compared.

The combination of regenerating, reheating and intercooling is shown in Figure 7. Different cycle arrangements are compared in Table I on the basis of the power requirements of each major element, with the capacity of the

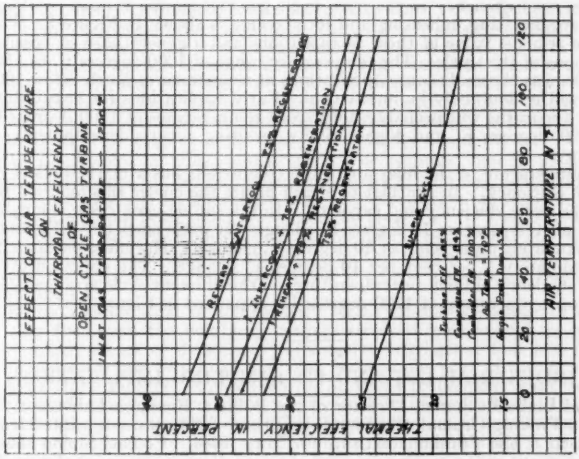


FIGURE 9.

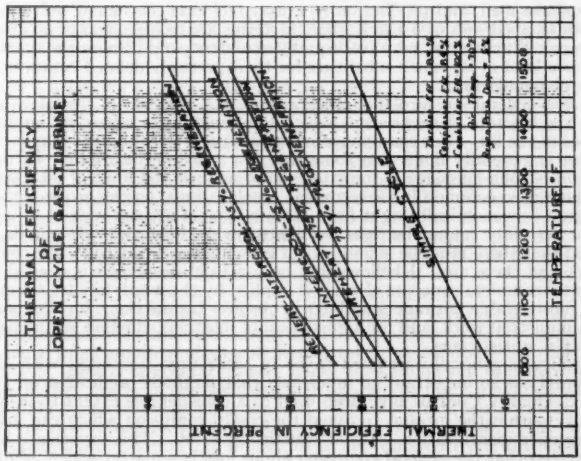


FIGURE 8.

generator as unity. The efficiency of each cycle arrangement at 1200 degrees F. is given, the temperature in degrees Fahrenheit, and the absolute pressure at different points in each cycle.

The thermal efficiency of the open cycle gas turbine for various combinations of regenerating, reheating and intercooling is shown in Figure 8. This gives the relative value of these modifications and a general idea of possible applications of this cycle. The temperature range of 1000 degrees F. to 1500 degrees F. was chosen because applications at temperatures much below 1000 degrees will undoubtedly be impractical. Early applications for heavy duty long life apparatus will probably not exceed temperatures of 1200 degrees F. with 1500 degrees F. and higher limits awaiting future developments in metallurgy.

The temperature of the inlet air to the compressor has a marked effect on the cycle efficiency. In contrast to steam plants the colder this inlet air the higher the cycle efficiency and capacity. In the simple open cycle, changing the inlet air temperature 10 degrees F. changes the cycle efficiency 0.74 points or approximately 3 per cent per 10 degrees F. change. The effect on capacity is approximately 4 per cent per 10 degrees F. change. The effect on cycle thermal efficiency of inlet air temperature, on the different cycle arrangements with a top temperature of 1200 degrees F., is plotted in Figure 9.

THE INDUSTRIAL LOT AND ITS SAMPLING IMPLICATIONS.

Colonel Leslie E. Simon, Director of the Ballistic Research Laboratory at Aberdeen Proving Ground, Maryland, is author of this paper reprinted from the Journal of the Franklin Institute for May, 1944. Colonel Simon shows that predictions of lot quality predicated on small samples are invalid in the absence of assurance of a knowable relation between the inspected sample and the uninspected remainder. Time-tried methods of making valid inferences from samples without increased cost are pointed out.

The General Concept of Lot. It may seem strange that such a well-known term—a term in such common usage—should merit special discussion. Engineers, business men, and production men may feel that they know quite well what a lot is, a sub-lot, or a batch, that the terms are flexible, and of no great importance. Statisticians may feel equally confident, and dismiss the matter at once as a kind of layman's substitute for their precisely defined concepts of universe or population.

Lot size is generally regarded as unimportant, if a property of the lot at all, yet it is the keynote of skepticisms which sometimes arise regarding implied properties of the lot. For example, a lot must have a size, and the maximum size of lot is generally limited in some way; e.g., by specifications. In this connection, strong language is sometimes used against the limit of 20,000 to a lot, which is so common in Government specifications, on the grounds that if the figure is economically right for a certain type of article, such as a fuze, it almost certainly cannot be right for almost everything else. It has been suspected that there must be sound logical grounds upon which lot-size could be based, and there are. I believe that only a brief discussion is needed to show that incomplete concepts of what is implied by the word, lot, are responsible for innumerable misunderstandings in business transactions, for economic losses in industrial work, and for a great deal of misdirected effort on the part of industrial statisticians.

Uniformity is generally assumed to be in some way associated with lot. This property is so important to munitions that methods predicated on the concept of Quality Control have been used to a considerable extent to insure uniformity within the lot and to avoid the costly confusion which can arise from improper identification of what constitutes a lot. This experience has demonstrated that the general confusion about lots is a matter which admits of correction quite readily, and in such a way as to remove both the existing

ills and losses and at the same time improve the quality of the products to which the clarified term is applied.

Webster's unabridged dictionary defines lot in the commercial sense as "an article, or parcel of articles, offered as an item for sale." As might be expected, this definition reflects rather accurately the generally accepted concept of lot. It is nothing more or less than an aggregation of articles. If the articles were produced more or less continuously by a well controlled mass production process, the lot may be a mere division with respect to time. There would be no harm in this loose and naive concept of lot were it not for the fact that a great many people who have not thought very seriously beyond this stage of the concept proceed to imply many more things about the lot.

The Implication of Homogeneous Lot. For example, lots of articles are generally accepted by the purchaser on a basis of sampling inspection. As soon as one speaks of samples there is an implied relation between the sample and the remaining unsampled portion of the large lot. If a relation did not exist, it would be futile to inspect the sample for the reason that no conclusion could be drawn with regard to the lot. In other words, one may simply fail to raise the question of the possibility of the total absence of homogeneity within the lot.

The tacit assumption of homogeneity seems especially strange in view of the fact that even the humblest sampler seems shrewd enough to have at least some concept of dispersion. His native intelligence tells him that no two articles are precisely alike and that perhaps one part of the lot may be different from another. Consequently, he tends to demand what he often calls a random sample, or at least a sample all articles of which do not come from the same portion of the lot. This second inference about the lot implies not only the relation between sample and lot, but further implies an ordered and at least knowable relation between different proportions of the lot. If the difference between articles sampled is too great, the purchaser may reject the lot on the grounds that the relation between its component articles is too loose even though the average of the articles sampled may be satisfactory. Strangely enough it does not occur to him to make a test to see whether there exists any relation at all between various portions of the lot.

Thus it is evident that, as used industrially and commercially, the term, lot, implies far more than a "parcel of articles." It must further imply that the lot is some sort of entity yet to be defined in which an ordered and knowable relation exists between the characteristics of portions of individuals which compose it.

The Statistician's Error. Whereas the layman's concept of lot or batch is much too loose, the statistician may make quite as grave an error in the opposite direction. The drawing of erroneous inferences from scant data is an anathema to him. He sees with crystal clarity the mote in his brother's eye, but let us for a moment examine the beam which he may be harboring in his own eye. When faced with a problem involving samples and lots the statistician is apt to identify the industrial lot with his own concept ofuniverse or population. Thus he may regard his problem as quite elementary and gayly proceed to apply his various statistical techniques, ranging from exact probabilities associated with multiples of the standard deviation in the normal distribution, to tests of randomness of sample. He as well as the layman may quite neglect to test for the existence of any universe at all.

The Importance of Order. As far as the single lot is concerned, often there is nothing that the statistician could do about it even if he realized his mistake. Given a small number of sample items, taken at random from an alleged universe, there is in general no way under the shining sun that he can determine if they are from one universe or from two or more universes which do not differ by an overwhelming amount. The difficulty exists for the greater part because of one reason only: because *order has been lost.* That one piece of essential information [2] would make the difference between almost no knowledge and a great deal of knowledge. If the samples had been taken in some purposive order; e.g., at approximately equal intervals with respect to

time during the production of the alleged lot, a great deal could be said about the singleness of universe and about the measures of its quality. However, it is also noteworthy that under these circumstances one would no longer be regarding the alleged lot as a lot, but rather making sub-lots from fractions of the alleged lot.

A Newly Expressed Definition of Lot. Let us examine briefly a concept of lot which can at least have a practical and useful purpose and have an operationally verifiable meaning. I venture a definition: *a lot is an aggregation of articles which are essentially alike.* That definition as it stands might satisfy the layman, but in order to make the definition operationally verifiable, some will demand that the phrase, essentially alike, be clarified. Let us then add: where essentially alike means that small sub-groups of sample items taken from the lot in arbitrary order will respond to the Shewhart [1] criterion of control.

It will be shown later that this definition of lot closely parallels the layman's vague and unvoiced concept of lot and the statistician's frank evaluation of its composition. It is hoped to show further that a general acceptance of this concept could achieve a large economic gain for our industrial economy. However, before doing so, let us examine briefly the manner in which a lot comes into being.

The Birth of a Lot. Let us for a moment consider a lot of piece parts made by a single automatic screw machine, where the diameter, X, of the piece part is an important quality characteristic. If there were no tool wear or other systematic or sudden changes in conditions, variations in diameter might be caused merely by slight random variations in the quality of flow of the cutting oil, minor variations in the raw material, slight changes in the ambient temperature of the room, and a large number of other causes having a small and variable effect. Since it is known that the combination of a large number of random variables results in a normally distributed variable, the quality characteristic, diameter of piece-parts, might under these conditions be normally distributed. However, these conditions do not obtain in practice. There is tool wear which progresses as a function of time and consequently the individuals of at least part of the distribution are not normally distributed about some mean value, $\bar{X}'$, where

$$\bar{X}' = \frac{\sum_{i=1}^{N} X_i}{N},$$

and N is the number of articles in the lot or universe of discourse. Instead, the individuals may be regarded as being distributed about a mean which is systematically shifting. Furthermore, the raw material is not entirely uniform but consists of discrete bars of stock whose quality characteristics with respect to machinability are likely to vary by discrete jumps from bar to bar. Therefore, the resultant lot of piece parts cannot be a smooth homogeneous universe or population of any describable form but instead its distribution consists of something like a broken line made up of short curves.

Now consider a lot of piece parts coming not from a single automatic screw machine but from several automatic screw machines. Even the initial settings of the machines could not have been exactly the same. Hence, there results a combination of sub-universes, each varying about a different mean value, $\bar{X}_1', \bar{X}_2', \cdots$. If the number of automatic screw machines were very large and the differences in the $\bar{X}'$'s small and randomly distributed, Mathematical Statistics tells us that the resultant distribution might again approach the well-known normal universe. However, in practice, the number of machines is not likely to be very large and the differences may be neither very small nor randomly distributed. In like manner, the variation associated with the respective machines might be different as measured by the standard deviations, $\sigma_1', \sigma_2', \cdots$, of the diameters of their respective products of piece-parts. Under

these circumstances one may certainly take a sample of n articles, compute the observed average, $\bar{X}$, where

$$\bar{X} = \frac{\sum_{i=1}^{n} X_i}{n},$$

and the sample standard deviation, sigma,

$$\sigma = \sqrt{\frac{\sum_{i=1}^{n} (\bar{X} - X_i)^2}{n}},$$

but certainly no statement can be made regarding the precise percentage of the lot which will be expected to fall within plus and minus any multiple of sigma.[1] That is a part of what Shewhart [1] means when he says that the 3-sigma limits in the criterion of control are not predicated on any value of probability but are so established because they work in practice.

The Shewhart Criterion of Control. The meaning of "working in practice" can be readily seen at this point. If none or very few of the $\bar{X}$'s, which came into being for a brief time, differ greatly one from another and if the series of standard deviations which existed from time to time, or from machine to machine, differ not too much one from another, then it is manifest that although one would have to regard the center of the distribution not as a mathematical line (without thickness), but rather as a band of finite width, and one would have to admit a degree of incertitude about the precise value of the standard deviation, nevertheless practically all of the distribution will lie within the average of any measure plus or minus 3 standard deviations of that measure. That is to say, practically all the individual X's will lie within $\bar{X}' \pm 3\sigma'$; and if random samples of size n are taken from the lot and the average, $\bar{X}$, of the sample and its standard deviation, σ, observed,[2] practically all $\bar{X}$'s will fall within $\bar{\bar{X}} \pm 3\sigma_{\bar{X}'}$ and practically all σ's will fall within $\bar{\sigma} \pm 3\sigma_{\sigma}$ where $\bar{\bar{X}}$ is the grand average of the $\bar{X}$'s, $\bar{\sigma}$ is the average of the σ's and σ_{σ} is the standard deviation of the σ's. This is well known from both theory and practice. That is to say, one could regard the lot as composed of members

[1] In this connection, Wilks [7], in a most outstanding article, has determined rigorously sample sizes necessary for setting tolerance limits to estimate the middle 100a % of the universe.

[2] Usually $\bar{\bar{X}}$ is taken as the true average $\bar{X}'$ where for k samples of size n,

$$\bar{\bar{X}} = \frac{\sum_{j=1}^{k} \bar{X}_j}{k} = \frac{\sum_{i=1}^{n} \sum_{j=1}^{k} X_{ij}}{nk},$$

Also, σ' may be taken as $\frac{\bar{\sigma}}{C_2}$ where

$$\bar{\sigma} = \frac{\sum_{j=1}^{k} \sigma_j}{k}$$

and C_2 is a factor to correct for the bias of sample size taken from the distribution of standard deviation.

$$\bar{\sigma} = \sqrt{\frac{2}{n}} \frac{\left(\frac{n-2}{2}\right)!}{\left(\frac{n-3}{2}\right)!} \sigma' = C_{2\sigma}\sigma'.$$

F. R. Helmert, *Astronomische Nachrichten*, Vol. 88, No. 2096, 122 (1876).
Tabulations of C_2 appear in references [1], [3], and [5].

$$\sigma_{\bar{X}} = \frac{\sigma'}{\sqrt{n}} \text{ and } \sigma_{\sigma} = \frac{\sigma'}{\sqrt{2n}}.$$

For practical purposes, however, similar treatment of the simple statistic, *range*, is often to be preferred (see Ref. [3], [5], and [6]).

which are essentially alike in that they differ one from another by little more than that which should be expected as random variations in a homogeneous universe.

It remains only to test whether these conditions existed in practice. This test is made with classic simplicity merely by taking small samples in arbitrary order (usually in order of production with respect to time) and testing to see whether the averages of the small samples fall within plus and minus three standard deviations of the grand average and whether the standard deviations of the small samples fall within plus and minus three standard deviations of the average standard deviation. If the condition is met, the arbitrarily chosen samples appear to behave as the randomly selected samples described in the preceding paragraph. It follows, then, that the lot appears to be homogeneous in the sense that discrete parts of the lot have the same characteristics as the lot as a whole (see Ref. [3], pp. 161–163). Thus, a meaning can be given to the term "essentially alike" which is subject to operational verification by the statistician and which is satisfactorily useful in industrial practice.

Lot-Size Inherent in the Production Process. Since it is so thoroughly evident that for practical purposes *the individuals of the lot must be essentially alike*, further discussion of the criterion of control can be deferred and some evident facts brought out at once. If the lot is to be a useful category, its size must be no greater than the number of articles which a manufacturer can make essentially alike. However, it is quite impossible to tell beforehand how long a manufacturer can produce articles which are essentially alike. This will vary not only from manufacturer to manufacturer but from time to time with the same manufacturer. Hence, it is quite impossible to prescribe *a priori* any lot-size. *The lot-size and the lot itself, with all the properties which describe its quality, are determined by the production process.*

Determination of Lots. In view of the observations made thus far, one may question if it is possible to identify and make up a lot. Whereas it is not proposed to treat the subject exhaustively, this question may be answered with the general statement that it is quite practicable to determine the varying quality of the continuous flow of the product as accurately as one chooses, and concomitant with this determination to break the continuous flow up into unit categories—call them lots or anything else—upon the demonstration of a change in "essential sameness" by as much or as little as is of economic consequence. The problem is solvable in an economic way with the expenditure of no more effort than that usually expended upon an invalid assessment of the quality of the current pseudo-lot.

Consider a series of categories differing with respect to size as follows: a grand-lot consisting of 10 to 40 lots, a lot consisting of 10 to 40 sub-lots, and a sub-lot consisting of 10 to 40 batches. Suppose further that it is possible to take small random samples (consisting of 4 to 40 articles) from the batches as they flow from the assembly or production line.

It is no more possible to judge the quality of the batch from the small random sample taken from that batch than it is to judge the so-called lot from the random sample taken from that lot. In neither case is there a check on the internal consistency of the unit collection sampled. Quality may have changed in the middle of the batch, and one does not know and cannot know whether such change has occurred. However, this may not be important for (1) the batch is a very small unit and the change, if any, is not likely to be great, (2) it will be shown subsequently that one can discover quickly a departure from essential sameness, if it occurs, and will know that it occurred at about batch number so-and-so, even though it will not be known whether the change occurred in the early, middle, or late part of the batch.

Suppose now that one observes the average value of the sample items from each batch expressed as the arithmetic mean, $\bar{x}$ (that is what is generally done), and one observes also the dispersion of the sample items of each batch expressed as standard deviation, σ; probable error, P. E. $= 0.6745\sigma$; or maximum dispersion, R, i.e., the greatest value minus the smallest value in the sample. Now, just a little statistical method must be added as a leaven to all

this common sense. It can be shown statistically that if all the batches are merely subdivisions of a larger, uniform, and consistent category, two things will occur:

(1) almost all the batch dispersions will fall within the range defined by the average of the batch dispersions ± a constant[3] times the average of the batch dispersions.

(2) almost all the batch averages will fall within the average of the batch averages ± another constant[3] times the average of the batch dispersions. This is essentially the Shewhart criterion of control mentioned in the earlier part of the discussion. Thus, the occurrence of a batch average or dispersion beyond these limits marks the end of a category within which *one is justified as acting as if all the articles were essentially alike.* Furthermore, one knows the quality of this larger category, the sub-lot, quite precisely, *for its average is very close to the average of the batch averages, and its dispersion is very close to a simple multiple of the average of the batch dispersions. Thus, without a very precise knowledge of batch quality, one has a very thorough knowledge of sub-lot quality.* This procedure answers the question of determination of quality only for a special set of circumstances.

The Purchaser's Dilemma. However, for the purchaser, this procedure is entirely impracticable. It is quite impossible for him to go into the manufacturing plant and sample the batches. In general, he cannot have access even to the sub-lots for sampling inspection, although, under some circumstances, this is practicable; e.g., under some Government contracts. Generally, the purchaser cannot specify the product and also govern its manufacturing process. Often he must be limited merely to accepting or rejecting the finished articles on a basis of a sampling inspection prescribed by the specification. Hence, the purchaser's samples are generally limited to the so-called lot, which may or may not be a lot in the true sense of the word, i.e., an aggregation of articles which are essentially alike. If the whole lot were available for the application of a planned sampling technique, the purchaser could, by purposive sampling, check on the homogeneity of the lot, but he could not tell where a change, if any, in essential sameness occurred because that precise piece of information, *order*, is lost in handling and shipment. The lot offered for acceptance, for example, may be composed of two or more sub-lots whose true levels differ significantly. Has, then, the purchaser any economic method of ascertaining the quality of the product offered by the vendor?

A Method of Quality Determination. There is an easy solution to this problem which under many circumstances may be almost good enough. Certainly the ideal method would be that of sampling the batches or sub-lots. Under those circumstances, there would be opportunity for locating the cause of poor quality when it occurred, and opportunity for eliminating the cause of the trouble, and thereby producing almost all good quality. That would be Process Quality Control. Whereas the door to Process Control is closed (1) by loss of order and (2) by the remoteness of the inspection from the cause of bad quality, nevertheless *quality determination* through the methods of Quality Control is entirely feasible.

It should be remembered that from the viewpoint of principle, it is not very important whether one samples the batch, sub-lot or lot. The basic principle still holds: *the valid quality prediction pertains to the echelon one order higher than the unit sampled.* If one samples sub-lots, valid quality predictions can be made about lots, for order has been preserved and a change in quality can be detected. If one samples lots, the valid quality prediction pertains to an aggregation of lots, or the grand-lot. The economic penalty simply is this: the larger the unit of valid quality prediction, the greater the quantity to be scrapped, reworked or one hundred per cent inspected when trouble occurs, and the greater the liability of failure to detect some minor or brief quality variations.

However, these penalties may be small. In very large-scale inspection for

[3] In practice, the mathematics of footnote[2] is generally reduced to this simple procedure.

acceptance, such as is now encountered in war production, altogether too much attention is liable to be focused on the occasional bad lot rather than on the whole group of sub-standard lots or on the bad manufacturer. If only the bad manufacturer is assuredly identified and the flow of his product arrested before it has done much harm, much will have been accomplished.

Illustration of True Lot Identification. In Table I is given the average, X, and the dispersion expressed as the range, R, for samples of five taken from 25 consecutive lots of a manufacturer of complete rounds in which the quality characteristic muzzle velocity[4] was measured. The lots which failed to meet the specification: $1715 \leq \bar{x} \leq 1785$, $R \leq 70$ are marked with an asterisk. This is the usual method of lot-by-lot acceptance inspection.

TABLE I

Lot	Mean Muzzle Velocity f/s	Range f/s
1*	1710	42
2*	1711	40
3*	1713	39
4	1718	26
5	1735	10
6	1739	25
7	1723	14
8	1741	15
9	1738	11
10	1725	31
11	1731	25
12	1721	19
13	1719	43
14	1735	39
15	1741	17
16	1783	51
17	1777	09
18*	1794	15
19	1773	37
20*	1789	54
21*	1798	15
22*	1789	29
23*	1788	39
24*	1799	30
25*	1807	44

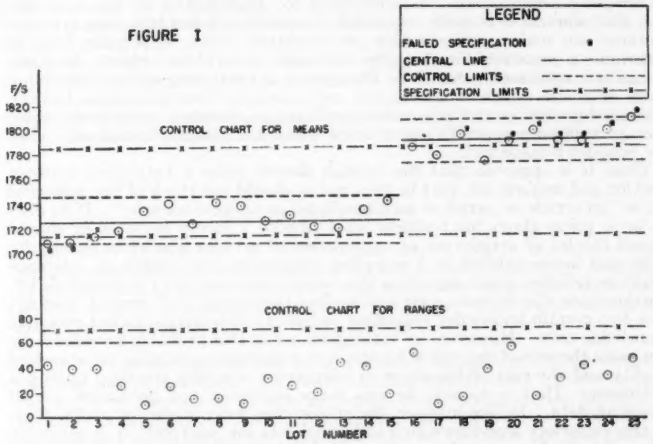
* Failed to meet specification.

In Figure 1 the Shewhart criterion of control is applied. It is quite evident that the first 15 lots are of essentially the same quality, and none should have been rejected. Those which were rejected (or retested until they passed) represent merely so much economic loss. However, near or between Lots 15 and 16 a change occurred. If the specification criteria are really important, all the Lots 16 through 25 (not just part of them) should have been rejected. Thus, it is all too evident that Lots 1 through 15 constitute the real lot (called the grand-lot) in the sense that they are *an aggregation of articles which are essentially alike.* In a like manner Lots 16 through 25 constitute another real lot, but a lot which should not pass the specification.

The Use of Attributes. In the above discussion attention has been focused entirely on measurement by variables, i.e., on quantitative measurements. It was pointed out that only those categories conforming to the Shewhart criterion of control should be combined to form the next higher homogeneous echelon. This may be compared, on the other hand, to measurement by

[4] The data have been altered in scale only so as to reveal no actual military information.

FIGURE I



attributes, i.e., qualitative measurements. In the case of attributes a homogeneous lot may be formed by mixing together thoroughly lots of different fraction defective, if mixing is physically practicable and provided that one is willing to accept the consequent result. For example, consider two lots, one of fraction defective $q_1 = 0.01$ and containing $N_2 = 1000$ articles, the other of fraction defective $q_2 = 0.04$ and containing $N_2 = 2000$ articles. By mixing thoroughly the two lots we may form a homogeneous lot of 3000 articles which will have a fraction defective,

$$q_3 = \frac{N_1 q_1 + N_2 q_2}{N_1 + N_2} = \frac{(0.01)\ (1000) + (0.04)\ (2000)}{3000} = 0.03.$$

In practice there may be some cases in which the resulting fraction defective is acceptable; in other cases, however, the final fraction defective may be intolerable.

Thus, whereas the consequences of occasional accepting a small quantity of material which is more defective than intended by the specification may be merely an adulteration of the flow of product in the sense that acceptance of N articles which are 3 per cent defective is equivalent only to 97 per cent of N articles which are perfect, nevertheless, it may be quite important to detect changes in the flow of a product measured by attributes and to classify portions of it by lots which contain known levels of fraction defective. The procedure for identification of lots of attributes is similar to that described for variables except that only one statistic (the fraction defective) is used and the sample size, n, should be somewhat larger. In order to be justified in regarding each increment of flow as essentially the same as its predecessors, the fraction defective of the sample from that increment should fall within

$$\bar{q} \pm 3\sigma_q \text{ where } \bar{q} = \frac{\sum_{i=1}^{k} q_i}{k},$$

and

$$\sigma_q = \sqrt{\frac{\bar{p}\bar{q}}{n}}, \quad \text{and} \quad \bar{p} = (1 - \bar{q}).$$

Other methods of setting limits are given in references [3], [5] and [6].

Sampling Implications. As a corollary to the definition of lot, it is seen also that whereas it is quite impossible to specify *a priori* how long a manufacturer can make articles which are essentially alike, it is quite easy to determine *a posteriori* how long he did make them alike. Hence, lot-sizes, in the layman's sense, should be thought of as pertaining to mere sampling units of a size which is convenient and economic (not as uniform units), whereas lot-sizes in the true sense can have no number preassigned, since they pertain to practically useful units which can be determined only after the product is inspected.

Thus, it is apparent that the layman should make a distinction between *unit* lot and *uniform* lot, that he does not or should not think of the industrial lot as "an article or parcel of articles offered as an item for sale." If he does so he is going about his business most blindly. Surely he must or should regard the lot of articles as an aggregation of articles which are essentially alike and hence subject to a sampling inspection from which an inference might be reliably drawn regarding the quality characteristics of the whole lot. Furthermore, the layman must realize that the sample from the lot does not give him certain knowledge regarding its quality characteristics but probable knowledge only. However, by intelligent use of this probable knowledge he can make the sum of the cost of his errors in mistakenly accepting sub-standard quality and the cost of his errors in mistakenly rejecting standard quality a minimum. That is to say, he can make economic and predictive use of sampling data. In like manner, the statistician must or should realize that, in this imperfect world in which industrial lots are produced, it is generally impossible for him to make precise probability statements about the lot, but that nevertheless he can make valid statements about the essential sameness of the articles which compose the lot which are entirely satisfactory for practical purposes.

Summary. Predictions of lot quality predicated on small samples are invalid in the absence of assurance of a knowable relation between the inspected sample and the uninspected remainder. Assurance of the existence of such a relation is readily obtained by taking small samples from arbitrary divisions of the alleged lot and then testing to see if the arbitrarily chosen samples appear to behave as random samples from a universe. The valid quality prediction then pertains to the echelon one order higher than the unit sampled. There are three degrees of efficiency in making quality predictions.

First, small random samples taken from the lot at an earlier time and at a different place may be inspected. Under this condition, the quality prediction is of very little value.

Second, the whole of the alleged lot may be available for inspection; it may be divided into arbitrary strata; sampled; and the criterion of control applied. Under these conditions, a knowledge of the validity or non-validity of the prediction is available, but no efficient corrective measures are possible if the lot appears to be non-homogeneous.

Third, small samples may be taken in order of production and the criterion of control applied. Under this condition valid lot-quality predictions are possible and efficient division of the flow of production into true lots is automatically obtained without additional cost.

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[3] L. E. Simon, "An Engineers' Manual of Statistical Methods," John Wiley & Sons, Inc., New York, 1941.

[4] American War Standard, Z1.1, Z1.2, "Guide for Quality Control and Control Chart Method of Analyzing Data," American Standards Association, New York, 1941.

[5] American War Standard, Z1.3, "Control Chart Method of Controlling Quality During Production," American Standards Association, New York, 1942.

[6] E. S. PEARSON, "The Application of Statistical Methods to Industrial Standardization and Quality Control," British Standards Institution, London, 1935.

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NOMOGRAMS.

Mr. Carl P. Nachod, of Nachod and United States Signal Company, demonstrates the great advantage of parallel line logarithmic type nomograms in the solution of many engineering problems. He shows how these useful devices may be constructed for given conditions. His article is reprinted from General Electric Review, May, 1944.

Parallel-line nomograms can be constructed to perform the operations of addition and subtraction, or of multiplication and division. Those designed for the latter purpose are of especially great value in engineering for they can be made to furnish immediately, and without any computation, the solutions of equations involving not only multiplication and division but also raising to powers and extracting roots. The present article will be confined to nomograms of this type.

To illustrate a typical application, assume that the problem in hand is to determine how many cubic feet of water would flow per second through a given size of pipe under the action of a given hydraulic head. If friction is neglected, these factors are expressed by the hydraulic equation $C = 6.3\, A^{\frac{1}{2}}\, B^{2}$; in which C is the cubic feet of water that will flow per second, A is the hydraulic head in feet, and B is the diameter of the pipe in feet. To solve this equation by mathematics would require four operations (the extracting the square root of one value, the squaring of another value, and two multiplications).

All these computing operations can be eliminated and much time saved by using the nomogram shown in Figure 1 which was constructed to fit the equation $C = 6.3\, A^{\frac{1}{2}}\, B^{2}$.

Thus, the answer to the assumed problem is read directly on the C scale at the point where this scale is crossed by a straight line between given A and B values on their respective scales. For instance, the dashed line shows that for a 9-foot head and a 2-foot diameter pipe the water flow would be 75.6 cubic feet per second.

How this particular nomogram was constructed will be described later in the article, along with the methods employed in constructing other parallel-line nomograms of the same general type.

UTILITY FOR DESIGNERS.

It is in the problem of working out the best engineering design, however, that nomograms have their greatest utility. In such work there are usually many different combinations of values that will fulfill the mathematical requirements but one of these combinations is likely to have the advantage of meeting the physical or economical requirements better than the others. By the use of a suitable nomogram and a movable straight line, the designer can immediately see what are the possible combinations of values and from them he can select the best combination for his purpose.

The straight-line mentioned is variously termed the computing secant, transversal, or index line. In this article it will be referred to simply as the secant. A straight line drawn on the underside of a transparent sheet makes a convenient form of movable secant.

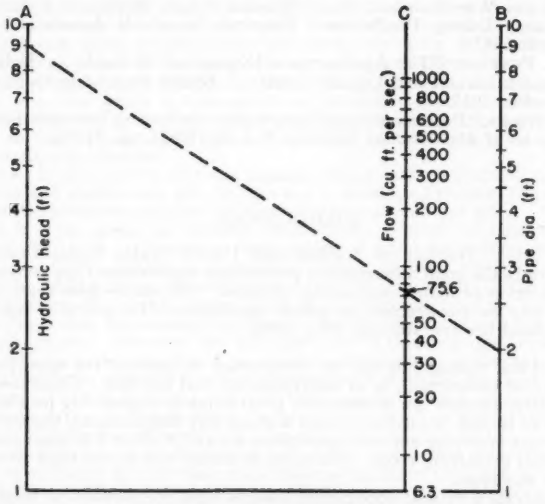

FIGURE 1.—A THREE-VARIABLE PARALLEL-LINE NOMOGRAM DESIGNED TO SOLVE PROBLEMS INVOLVING THE HYDRAULIC EQUATION $C = 6.3\ A^{\frac{1}{2}}\ B^{\frac{3}{2}}$. FOR SIMPLICITY OF ILLUSTRATION, ONLY THE MAIN GRADUATIONS ARE HERE SHOWN. INTERMEDIATE GRADUATIONS WOULD BE INCLUDED IF NEEDED.

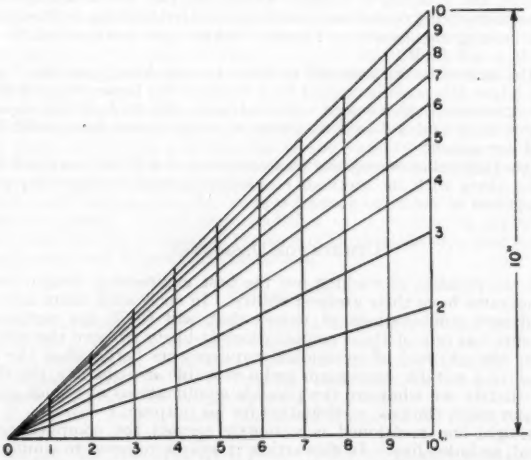

FIGURE 2.—ILLUSTRATION OF A DRAWING THAT SERVES AS A CONVENIENT MEANS FOR GRADUATING NOMOGRAM SCALES.

For Equations Having Three Variables.

For solving an equation that can be expressed in one of the following forms:

$$C = AB \qquad C = \frac{A}{B} \qquad C = \frac{1}{AB}$$

in which A, B, and C are variables having any fractional or intregral exponents, use is made of a nomogram having three scales—one scale for each variable. By suitable scale positioning, the nomogram can be adjusted to compensate for the inclusion also of multiplying constants in the equation. The nomogram in Figure 1 is of this type, because of the presence of the multiplying factor 6.3 in its equation $C = 6.3\ A^{\frac{1}{3}}\ B^{\frac{2}{3}}$.

For Equations Having More Than Three Variables.

Equations that contain more than three variables of the character defined in the preceding paragraph can be readily solved by suitable combining operations—provided the equations do not require that any of the variables be added to or subtracted from each other. The following (with the option of including multiplying constants) are typical forms of four-variable equations for which parallel-line nomograms can be prepared.

$$C = ABD \qquad C = \frac{AB}{D} \qquad C = \frac{A}{BD} \qquad C = \frac{1}{ABD}$$

For each variable over three, there is required one extra nomogram scale and one extra secant, as will be described later.

Principle of the Nomograms.

Parallel-line nomograms of the type described in this article have scales which are logarithmic and which are so positioned as to utilize variations of the principle that the sum of the logs of numbers gives the log of the product of the numbers. The addition or subtraction of the logs is automatically done by the placing of the secant across the scales. Logarithmic graduations have two other advantages: they have the same percentage error at any point on the scale, and they compress a great range into a short length.

The relative positions of the scales conform to requirements of the particular equation which the nomogram represents. The same is true of the length of each scale between graduations 1 and 10 (or between 10 and 100, etc., which is of course the same since the scales are lograithmic). Whatever this length may be, it is termed the scale *modulus*, and is represented by the symbol m.

The moduli of the scales and also the direction in which the scale graduations progress (whether upward or downward) also conform to relationships present in the equation.

Since the method used in graduating the scales is the same regardless of the positions, directions, or moduli of the scales, the means of performing that operation will be described next.

Graduating the Scales.

In some instances the graduating of the scales may be done by direct transfer of the graduations from logarithmic cross-section paper or from either the AB or CD scales on a slide rule. Usually, however, a graduating means such as that shown in Figure 2 will be needed. Once made, it can be preserved for repeated use; and it will be found of great convenience because it provides an easy means of graduating scales of different moduli.

The drwaing in Figure 2 is laid out on cross-section paper having web lines that are uniformly spaced and reasonably close together. (These web lines are not shown in Figure 2.) Preferably the drawing should be about as large as 10 inches wide and 10 inches high, the exact size being determined by convenience in utilizing the web lines of the paper. When made that size,

the drawing will cover the usual range of requirements for scale graduating, since usual moduli may vary from 2 to 10 inches.

The horizontal scale is then marked in ten equal divisions; and the right-hand vertical scale is laid out logarithmically with the aid of a table of logarithms, or if of the same length as the CD scales of a slide rule it can be transferred from one of these scales. (This work of laying out the vertical scale logarithmically could of course be eliminated by making the drawing on so-called "semi-log" cross-section paper* provided such paper is available with a modulus about 10 inches long.)

The addition of sloping lines, such as shown in Figure 2, completes the drawing; and it is ready for use in graduating nomogram scales of any desired modulus.

Constructing Three-variable Nomograms.

The method of constructing three-variable parallel-line nomograms will now be described, as applied to the typical nomogram of this type shown in Figure 1.

The initial step is to determine the spacing between the scales. This step is taken after first disregarding all multiplying constants and all exponents of the variables in the equation which the nomogram is to match. Thus the equation $C = 6.3\ A^{\frac{1}{2}}\ B^{2}$ (for Figure 1) is to be considered initially as $C = AB$. For the scales A and B draw two parallel vertical lines, say 10 inches apart, and at the bottom of the paper draw a horizontal base line for the start of the graduations, as shown in Figure 3.

Assume some length for the modulus of each outer scale, as for instance 20 inches for A and 5 inches for B. Lay off these distances upward from the base line, and from their tops draw diagonals to the base points of the opposite scales. The intersection of the diagonals determines the location for scale C which is now drawn parallel to the other scales. The height of this intersection point above the base line will be the modulus for scale C, and is found to be 4 inches as indicated in Figure 3.

Incidentally, it may be noted in Figure 3 that scale C is so located as to divide the distance between scales A and B in the ratio of their moduli, i.e., as 20:5 or 4:1, since AC is 8 inches and CB is 2 inches, AB being 10 inches.

Introducing the previously disregarded exponents in equation $C = 6.3\ A^{\frac{1}{2}}\ B^{2}$ has the effect of multiplying the moduli of scales A and B by those exponents. Thus, the modulus of scale A will be halved, making it 10 inches; and that of B will be doubled, making it also 10 inches. This coincidence of 10 inches as the final modulus of both these scales resulted from the choice of 20 inches and 5 inches as their initial moduli. A choice of initial moduli having other than a 4:1 ratio would in this case have resulted in the A and B scales having unequal final moduli; and this condition would increase the work of graduating the scales for their graduations would not be alike. However, the nomogram would be equally valid. It should be noted that too great a ratio of the moduli of the A and B scales, say more than 5 or 6:1, will place the middle scale so close to an outer scale as to reduce the graphical accuracy. Whether the final moduli of the A and B scales are equal or unequal, each is to be marked 10 at its top and 1 at its base.

To arrange for including in the nomogram the hitherto disregarded constant 6.3 in the original equation $C = 6.3\ A^{\frac{1}{2}}\ B^{2}$, solve the equation for $A = 1$ and $B = 1$. The result is $C = 6.3$. On the unfinished nomogram, Figure 3, this operation is equivalent to drawing a secant between $A = 1$ and $B = 1$, and then finding that the point where the secant crosses the C scale (at its base) is to be marked 6.3. It will be noted that this operation of including the equation constant may also be regarded as moving the C scale downward so as to start with 6.3 at its base. Incidentally, it may be noted that the point where the construction diagonals intersected has now become 63 for, as has been stated, this point is one C scale modulus above the base of that scale.

* Semi-log cross-section paper has uniformly spaced vertical lines and logarithmically spaced orizontal lines.

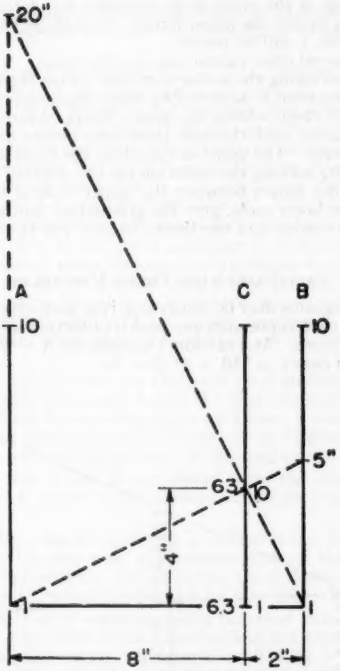

FIGURE 3.—ILLUSTRATION OF THE METHOD EMPLOYED IN CONSTRUCTING THE NOMOGRAM SHOWN IN FIGURE 1.

The three scales being now positioned by at least one significant graduation point on each, there remains only the operation of graduating them throughout, increasing upwards, within the limits of the paper. This is done by transferring to the A and B scales (which are of 10-inch modulus) the graduations obtained from the 10-inch long vertical line of the drawing represented in Figure 2; and, similarly, transferring to the C scale (which is of 4-inch modulus) the graduations obtained from the 4-inch long vertical line in the drawing. (Had the moduli been other than 10 inches and 4 inches, the scale graduating would be done with the aid of vertical lines of corresponding length in Figure 2, drawing them in for the purpose if not already there.)

The nomogram thus completed and ready for use was shown in Figure 1.

From the foregoing description relating to Figure 3, the following general features of three-variable nomogram construction are apparent. The relative positions of the outer scales and the choice of initial moduli for these scales automatically determine the position and modulus of the inner scale. The final modulus of each outer scale is determined by multiplying its initial modulus by the exponent of the variable which the scale represents.

The graduations of the outer scales start with 1 at the base line. Those of the inner scale start at the base line with a value equal to the multiplying constant in the equation if there is such a constant; otherwise these graduations start with 1, like those of the other scales.

The required range of the outer-scale variables will determine the choice of the moduli for them to suit the paper limits. In making the choice, the drawing pictured in Figure 2 will be useful.

The following general observation can also be made. A different and very practical way of positioning the scales vertically (after their moduli have been chosen and their horizontal separations have been determined) is first to graduate any two of them within the paper limits, arbitrarily select a value on each of these scales, and through these two points draw a straight line crossing the third scale. The point so found on the third scale will then have the value obtained by solving the equation for the selected values of the other two variables. A line drawn between the upper ends of the two scales, and another between the lower ends, give the graduation limits of the third scale. With this significant point and the limits known, the third scale can now be graduated.

Variations with Three Variables.

Parallel-line nomograms may be constructed for such expressions as $C = AB$ even when A and B each represents any positive function or quantity containing one variable but no more. As a random example, let $A = 10 + a$, $B = 6 - b$, and $C = c$ in which case $c = (10 + a)(6 - b)$.

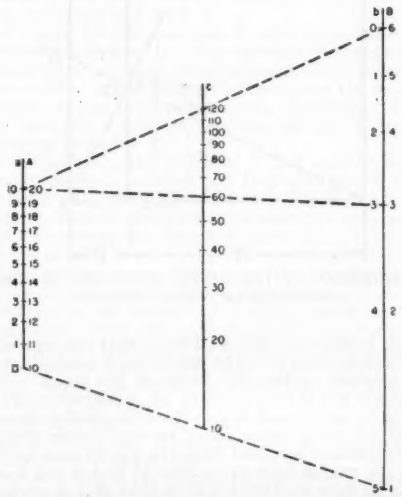

FIGURE 4.—A NOMOGRAM CONSTRUCTED TO FIT A CASE WHERE TWO OF THE VARIABLES EACH INCLUDES A CONSTANT: AS $A = 10 + a$ AND $B = 6 - b$. THE NOMOGRAM HAVING BEEN CONSTRUCTED, THE A AND B SCALE GRADUATIONS CAN BE ERASED.

A convenient and practical way of starting the construction of an appropriate nomogram is to draw a vertical line near each edge of the sheet for scales A and B, as in Figure 4. Then, if the same value of modulus m is chosen for both the A and B scales, the position of the c scale will be midway between the A and B scales, as shown also in Figure 4. That the c scale will be in mid-position can readily be noted by referring back to Figure 3 and

observing that the dotted construction diagonals (which locate the position of scale c) would intersect midway between scales A and B if those scales had equal moduli.

Next assume a series of successive values of A, as from 10 to 20, and from the equation $A = 10 + a$ determine the corresponding values of a. On one side of vertical-line A in Figure 4 lay out this 10-to-20 range of A values, using as the modulus for the graduations the modulus length that had been chosen. On the other side of this scale mark the corresponding values of a. Apply the same procedure to any convenient portion of vertical-line B, using a range of B values say from 1 to 6. In graduating the B scale, the same modulus length is used as was used for scale A. On the other side of the B scale mark the corresponding b values, as determined from equation $B = 6 - b$.

For $A = 10$ and $B = 1$, the given equation $C = AB$ shows $c = 10$ which is located as a significant point on scale c by a straight line between the bottom ends of scales A and B. A straight line across the top end of these scales (which have values $A = 20$ and $B = 6$) locates on scale c the significant point $c = 120$, as determined also from the given equation $C = AB$. Now that at least one significant scale point is located on vertical-line c, the c scale can be graduated.

The graduating will be done on the basis of a modulus which is one-half as long as that chosen for scales A and B. The verification of this relationship can be visualized by referring again to Figure 3 and noting that, if the dotted construction diagonals started from equal heights on A and B, they would intersect at half that height on a C line midway between A and B.

Since the quantities A and B are absent from the equation $c = (10 + a)(6 - b)$, the A and B scales of the nomogram are no longer needed and may be erased.

That the nomogram fits the equation is indicated not only by the construction secants across the top ends and bottom ends of the scales but also by readings of any other secant. For instance, the random secant from $a = 10$ to $b = 3$ gives $c = 60$, which is checked by $c = (10 + 10)(6 - 3) = 60$.

The foregoing method of construction is practical only when the functions are quite simple. Note that if the equation had not been factored as it was, but had been multiplied out, in the form $c = 60 + 6a - 10b - ab$, it would not be suitable for this type of nomogram construction.

Also, it is to be noted that the base line of this nomogram is not shown, for it would lie below the bottom boundary of Figure 4. Furthermore, it would not be horizontal, as were those in Figures 1 and 3, but would start at the point $B = 1$ and slope downward to the left. This is one instance in which a sloping base line will result in a more compact nomogram. Another instance, under different circumstances, will be pointed out later in connection with Figure 7.

When the Equation Is Not Known.

A nomogram can be constructed for an equation even when the equation contains some function whose values must be determined from an empirical curve. In such a case, a set of abscissa values of the empirical curve are graduated on one side of the related nomogram scale and the corresponding empirical ordinates on the other side, and then the set that is not needed is erased. This elimination of scale markings no longer needed is similar to the suggested erasure of the A and B values in Figure 4.

A nomogram that involved the use of empirical-curve values in its construction is shown in Figure 5. In this particular case, use had to be made of *two* empirical curves but this meant merely duplicating for each of them the operations described in the preceding paragraph. The two empirical curves are shown in Figure 6. Their abscissa and ordinate values were laid out on the center scale of the Figure 5 nomogram, following which the no-longer-needed C_B values were erased, leaving only the two sets of l_0/r values.

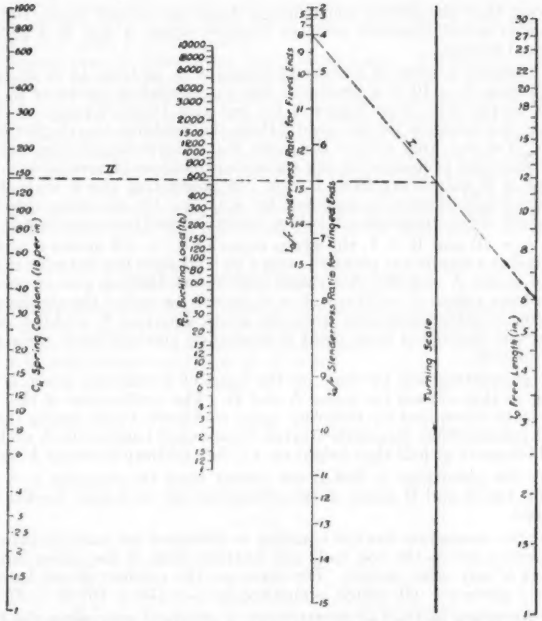

FIGURE 5.—A NOMOGRAM THAT GIVES THE BUCKLING LOAD OF HELICAL COMPRESSION SPRINGS AND THAT IS PRESENTED HERE AS AN EXAMPLE OF A NOMOGRAM THAT HAS SCALES (THE TWO l_0/r SCALES) WHICH ARE GRADUATED FROM EMPIRICAL SCALE VALUES.

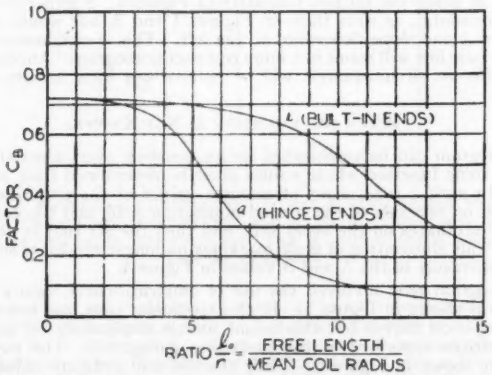

FIGURE 6.—EMPIRICAL CURVES THAT WERE UTILIZED IN GRADUATING l_0/r SCALES IN FIGURE 5.

Other Variations of Three-variable Nomograms.

It may be desired that the nomogram for the equation $C = AB$ have its C scale in an outside position, which means that the equation should be rewritten either as $A = C \times 1/B$ or as $B = C \times 1/A$. In either case, the same manner of subsequent procedure will be followed.

For the equation having the form $A = C \times 1/B$, the outer scales will be C and B; but B will have an inverse graduation, increasing downward. The reason for progressing downward is that this quantity appears in the denominator of the equation, hence varies inversely with A and C and therefore its scale must progress in the direction opposite to that of scales A and C.

For such a case the base line is best inclined, as a reference to Figure 7 shows that this feature of construction will make the nomogram more compact. On the completed nomogram, however, this base line will not be shown because having served the purpose of construction it would no longer be needed.

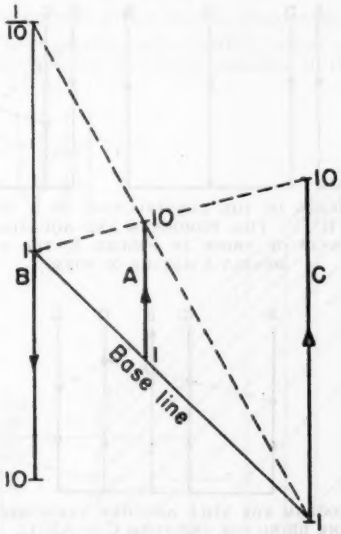

Figure 7.—Constructional features of a nomogram having a sloping base line.

The position and modulus of scale A are determined in Figure 7 by the intersection of the dotted constructional diagonals, which are drawn by fundamentally the same procedure as was outlined for Figure 3. It is to be noted, however, that while one of the diagonals from B starts from 1, the other starts from 1/10 instead of 10 to cause the diagonals to intersect between scales B and C.

Constructing Four-variable Nomograms.

In equations having four variables, for example $AB = CD$, let $AB = Q$ and $CD = Q$ as represented by the charts at the left and center in Figure 8. By having given Q the same modulus in each, the two charts may be superposed so as to make Q common to both. With the secants of each intersecting

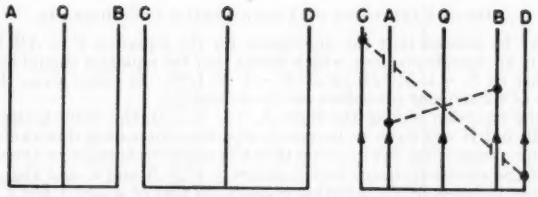

FIGURE 8.—CONSTRUCTIONAL FEATURES OF A NOMOGRAM FOR A FOUR-VARIABLE EQUATION OF THE FORM $AB = CD$. THE LEFT AND CENTER CHARTS (IN WHICH Q IS INTRODUCED AS A COMMON RELATING FACTOR) ARE SHOWN SUPERPOSED AT THE RIGHT TOGETHER WITH TWO SECANTS.

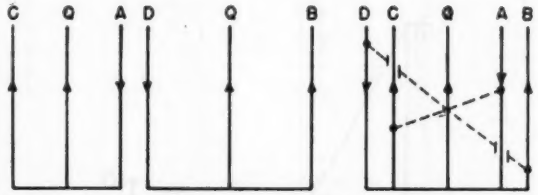

FIGURE 9.—ELEMENTS IN THE CONSTRUCTION OF A NOMOGRAM FOR THE EQUATION $C/A = B/D$. THIS NOMOGRAM AND EQUATION ARE THE MATHEMATICAL EQUIVALENTS OF THOSE IN FIGURE 8, THE DIFFERENCES BEING MERELY A MATTER OF FORM.

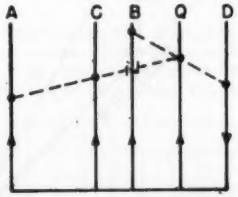

FIGURE 10.—NOMOGRAM FOR STILL ANOTHER VARIATION OF THE EQUATION $AB = CD$, THIS ONE BEING FOR THE FORM $C = AB/D$. IT IS CONSTRUCTED BY THE SAME GENERAL METHOD AS DEPICTED IN FIGURES 8 AND 9.

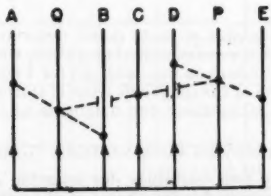

FIGURE 11.—COMPONENTS OF A NOMOGRAM FOR THE FIVE-VARIABLE EQUATION $C = ABDE$. TWO TURNING SCALES, Q AND P, AND THREE SECANTS ARE USED.

at the same point on Q, as in the right-hand chart of Figure 8, the Q value is algebraically eliminated and no scale graduations are required on scale Q. This scale is known as a turning or pivot scale.

With the scales spaced and the moduli determined, a set of values for three of the variables may be assumed and the fourth value calculated from the equation. The secant through the third variable and the intersection on Q determines a significant point for graduating the fourth scale. In Figure 8 and in subsequent Figures, the secants are broken where no intersections are intended.)

If the equation is solved as $C/A = B/D$, let $C/A = Q$ and $B/D = Q$. Then A and D are inverse and their scales will increase downward, as shown in Figure 9 together with the superposing operation similar to that in Figure 8.

If the equation is solved as $C = AB/D$, let $B/D = Q$. Then $C = AQ$ as shown in Figure 10, in which D increases downward. A completed nomogram of this general type was shown in Figure 5, except there the conditions required that the B and D scales increase in the direction opposite to those in Figure 10.

Constructing Five-variable Nomograms.

If an equation has five variables, as $C = ABDE$, let $AB = Q$ and $DE = P$. Then $C = QP$. Since there are no inverse variables, all the scales increase

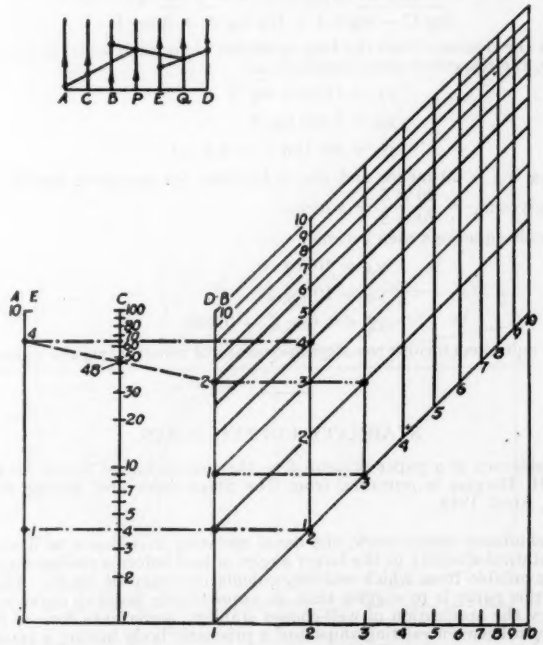

Figure 12.—The small inserted nomogram for the equation $C = ABDE$ is of the general type of that shown in Figure 11. The network nomogram, though more complicated to construct, enables solutions to be read with the use of only one secant.

upward; and Q and P are ungraduated turning scales as in Figure 11. As will be noted, this nomogram requires the use of three computing secants.

Increasing familiarity with the principles of nomogram design will at times reveal how a departure from conventional design may yield a nomogram that will save even more time than is normally saved by these graphic computing devices. Such cases require individual consideration but the following one of them will be referred to here as an example.

It concerns the five-variable equation C = ABDE, for which a conventional nomogram requiring five completely graduated scales and three secants could be laid out according to the insert diagram in Figure 12. The network variation of design, also shown in Figure 12, enables solutions of the equation to be obtained through the use of only one secant. Thus, such problems as "What would be the cost of a piece of material so long, so wide, and so thick, at so much per cubic inch?" can be solved with the minimum of time and effort by the network nomogram shown.

Algebraic Analysis.

For those who may wish to note the algebraic relationships that parallel the geometric construction of a nomogram, reference may be made to the initial equation $C = 6.3\, A^{\frac{1}{2}}\, B^{2}$ and its nomogram in Figures 1 and 3. The equation may be rewritten as $C/6.3 = A^{\frac{1}{2}}\, B^{2}$. Passing to logs,

$$\log C - \log 6.3 = 1/2 \log A + 2 \log B$$

If y is the distance from the base in inches along each scale, and m is the modulus, the respective scale equations are

$$y_A = 1/2\ m_A \log A$$

$$y_B = 2\ m_B \log B$$

$$y_C = m_C (\log C - \log 6.3)$$

Letting $m_A = 20$ inches and $m_B = 5$ inches (as was done earlier in the article) gives $m_C = \dfrac{20 \times 5}{20 + 5} = 4$ inches.

The scale equations then become:

$$y_A = 10 \log A$$

$$y_B = 10 \log B$$

$$y_C = 4 (\log C - 0.799)$$

These equations furnish the algebraic means of constructing the nomogram.

STABILITY COEFFICIENTS.

This abstract of a paper presented to the Institution of Naval Architects by N. H. Burgess is reprinted from The Shipbuilder and Marine Engine-Builder, April, 1944.

In preliminary design work, the naval architect may desire to have some idea of statical stability at the larger angles of heel before a preliminary body plan is available from which stability calculations may be made. The purpose of this paper is to suggest than an approximate stability curve may be drawn by the application of well-chosen stability coefficients derived from a comparison between existing ships and a prismatic body having a transverse section the same as the midship section of the design, the depth of the prism being equal to the depth of ship to the freeboard deck plus one-third of the mean sheer. The method proposed is to calculate the movement of the center of buoyancy of the prism at every 10 degrees from the upright position to

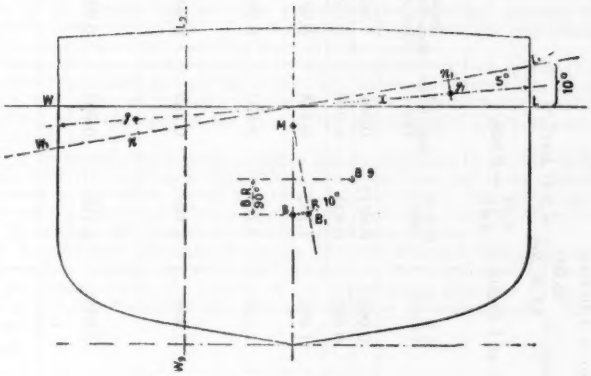

FIGURE 1.

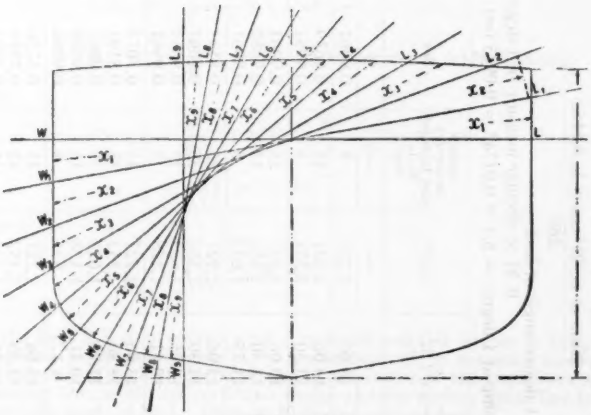

FIGURE 2.

TABLE I.—CALCULATION OF B R FOR PRISM CORRESPONDING TO A WHALER.

B = 26.03 feet. D = Moulded depth plus one-third mean sheer = 15.953 feet.

Immersed sectional-area of prism = 268 square feet.

B R of prism at angle of heel = $\frac{[\text{Sum of } (x/2)^3 \cos\theta] \times 0.1163}{268}$

B R at 1 degree angle of heel = B M × circular measure of 1 degree.

B R of ship at 1 degree = 5.3 × 0.01745 = 0.0925 feet.

B R of prism at 1 degree = 5.47 × 0.01745 = 0.0955 feet.

B M of ship = 5.30 feet.

B M of prism = $\frac{26.03^3}{12 \times 268}$ = 5.47 feet.

$\frac{\text{Ship}}{\text{Prism}}$ ratio at 1 degree = $\frac{5.30}{5.47}$ = 0.968.

Angle of Heel, deg.	x/2 ft.	$(x/2)^3$	Angle θ between x and Inclined W. L., deg.	$\cos\theta$	$(x/2)^3 \cos\theta$	Sum of $(x/2)^3 \cos\theta$	B R for Prism.	B R for Ship.	Actual Ship/Prism Ratios.	Basic Ship/Prism Ratios.
1	—	—	—	—	—	—	0.0955	0.0925	0.968	1.000
10	13.06	2,227	5	0.9961	2,216	2,216	0.962	0.931	0.968	0.997
20	13.06	2,227	15	0.9659	2,150	4,491	1.951	1.732	0.888	0.913
	13.30	2,352	5	0.9961	2,341					
30	13.06	2,227	25	0.9063	2,018	5,904	2.562	2.260	0.882	0.904
	13.30	2,352	15	0.9659	2,271					
	11.75	1,622	5	0.9961	1,615					
40	13.06	2,227	35	0.8191	1,822	6,642	2.885	2.565	0.889	0.907
	13.30	2,352	25	0.9063	2,130					
	11.75	1,622	15	0.9659	1,567					
	10.41	1,128	5	0.9961	1,123					
50	13.06	2,227	45	0.7071	1,573	6,890	2.995	2.710	0.905	0.920
	13.30	2,352	35	0.8191	1,925					
	11.75	1,622	25	0.9063	1,470					
	10.41	1,128	15	0.9659	1,090					
	9.42	835	5	0.9961	832					
60	13.06	2,227	55	0.5736	1,277	6,744	2.930	2.735	0.934	0.945
	13.30	2,352	45	0.7071	1,662					
	11.75	1,622	35	0.8191	1,328					
	10.41	1,128	25	0.9063	1,022					
	9.42	835	15	0.9659	807					
	8.67	651	5	0.9961	648					

90 degrees of heel by the method of in-and-out wedges. The method is simple and can be quickly applied without the use of an integrator.

The object of the calculation is to find the moment of transferred buoyancy of the emerged wedge when moved to the immersed position on the opposite side of the prism. This moment is taken about the axis formed by the intersection of the bounding lines of these wedges.

Water-lines are drawn at every 10 degrees of heel in such a way that the immersed sectional area of the prism is constant throughout the 90 degree range, the height of the water-line in the upright position being equal to the moulded mean draught of the ship in the condition for which the stability curve is required.

The immersed and emerged wedges will be identical up to the deck-edge angle when the section is wall-sided; the water-lines for these wedges will intersect on the center-line of the prism and form similar triangles (Figure 1). When the inclined water-line has passed the deck edge, the water-lines bounding the wedges will intersect at various positions (Figure 2).

It is sufficiently accurate to assume that the emerged wedge at every 10 degrees is a triangle approximately similar to the corresponding immersed wedge; consequently, the lengths of the dotted angle bisectors x_1, x_2, x_3 . . . can be used to determine the area and moment of transference. Thus, in Figure 1:—

Let

$$a = \text{area of wedge},$$

$$A = \text{immersed sectional-area of prism};$$

then,

$$\frac{a \times g\, g_1}{A} = B\, B_1, \text{ where } B\, B_1 \text{ is parallel to } g\, g_1.$$

Now,

$$a = \frac{x}{2} \times \frac{x}{4} \times 0.1745$$

$$= \frac{x^2}{8} \times 0.1745 \text{ (as 10 degrees} = 0.1745 \text{ radian)},$$

and

$$a \times g\, g_1 = \frac{x^2}{8} \times 0.1745 \times \frac{2x}{3} = \frac{x^3}{8} \times 0.1163$$

$$= \left(\frac{x}{2}\right)^3 \times 0.1163.$$

Hence,

$$B\, B_1 = \frac{\left(\frac{x}{2}\right)^3 \times 0.1163}{A}.$$

The lever B R is, of course, what is required, and this is equal to $B\, B_1 \cos \theta$. Therefore, in the summation of the moments it will be necessary to multiply the moment of each 10-degree wedge by the cosine of the angle between the bisecting line, x, of that particular wedge and the surface water-line for any particular angle of heel. Thus, at 30-degree angle of heel:—

Moment of

First 10-degree wedge is multiplied by cos 25 degrees,
Second 10-degree wedge is multiplied by cos 15 degrees,
Third 10-degree wedge is multiplied by cos 5 degrees.

TABLE II.

Index No.	Type of Vessel.	Breadth Mld. on W. L.	Mean Draught.	Freeboard plus 1/3 Mean Sheer.	Depth plus 1/3 Mean Sheer.	Displacement Coefficients.			Topside Coefficient.	$\frac{K B}{d}$ (Ship).	$\frac{K B}{d}$ (Prism).
						Block.	Midship Section.	Prismatic.			
1	Trawler.....	22.50	11.05	3.585	14.635	0.540	0.780	0.693	0.830	0.6075	0.594
		22.50	9.00	5.635	14.635	0.478	0.736	0.650	0.810	0.618	0.606
2	Cruiser.....	67.06	22.25	11.750	34.000	0.576	0.858	0.672	—	0.572	0.544
		64.00	14.46	19.540	34.000	0.523	0.827	0.633	—	0.572	0.548
3	Destroyer.....	31.00	11.81	8.850	20.660	0.521	0.775	0.672	0.739	0.597	0.575
		29.56	7.00	13.660	20.660	0.439	0.716	0.613	0.719	0.627	0.600
4	Tug.....	33.00	12.00	5.400	17.400	0.533	0.914	0.584	0.796	0.584	0.536
		33.00	10.00	7.400	17.400	0.498	0.897	0.556	0.772	0.589	0.541
5	Cruiser.....	63.40	22.25	11.750	34.000	0.600	0.890	0.674	0.771	0.565	0.537
		61.15	14.46	19.540	34.000	0.543	0.866	0.628	0.756	0.569	0.540
6	Cargo vessel...	61.17	25.29	13.710	39.000	0.717	0.938	0.765	0.852	0.542	0.522
		59.85	13.00	26.000	39.000	0.656	0.904	0.726	0.823	0.549	0.522
7	Whaler.....	28.00	14.00	4.060	18.060	0.609	0.896	0.680	0.880	0.591	0.546
		28.00	10.00	8.060	18.060	0.533	0.857	0.622	0.842	0.589	0.563
8	Tanker.....	70.00	31.70	10.000	41.700	0.789	0.986	0.800	—	0.527	0.506
		70.00	19.62	22.080	41.700	0.736	0.976	0.754	—	0.531	0.510
9	Channel steamship...	46.00	12.50	13.690	26.190	0.540	0.908	0.595	0.802	0.576	0.534
		45.87	9.00	17.190	26.190	0.491	0.878	0.559	0.781	0.577	0.546
10	Whaler.....	26.03	12.00	3.950	15.950	0.538	0.858	0.627	0.819	0.589	0.560
		25.93	8.50	7.450	15.950	0.471	0.802	0.587	0.782	0.595	0.579
11	Cargo vessel...	56.00	23.33	12.590	35.920	0.724	0.934	0.775	0.846	0.546	0.523
		54.70	11.83	24.090	35.920	0.665	0.896	0.743	0.816	0.548	0.527
12	Trawler.....	23.33	11.50	3.670	15.170	0.575	0.802	0.717	0.825	0.600	0.585
		23.33	8.50	6.670	15.170	0.504	0.735	0.686	0.802	0.612	0.609
13	Depot.....	85.00	26.00	17.150	43.150	0.619	0.949	0.652	0.811	0.554	0.521
		85.00	14.09	29.060	43.150	0.536	0.902	0.595	0.779	0.556	0.538
14	Channel steamship...	39.33	12.00	12.920	24.920	0.561	0.962	0.583	0.822	0.565	0.515
		39.33	8.50	16.420	24.920	0.506	0.947	0.534	0.798	0.571	0.521
15	Destroyer.....	32.70	10.92	8.420	19.340	0.586	0.868	0.675	—	0.589	0.556
		32.98	8.58	10.760	19.340	0.533	0.831	0.642	—	0.597	0.567
16	Cargo vessel...	57.67	25.02	13.980	39.000	0.780	0.990	0.788	—	0.532	0.506
		57.67	15.62	23.380	39.000	0.749	0.983	0.762	—	0.538	0.508

16	Cargo vessel...	57.67	15.62	23.380	39.000	0.749	0.983	0.762	—	0.538	0.508

TABLE II.—(*Continued.*)

Index No.	Type of Vessel.	Ship/Prism B R Ratios. Angles of Inclination in Degrees.										B M Ratio, Ship/Prism.
		1	10	20	30	40	50	60	70	80	90	
1	Trawler.....	1.00	0.990	0.915	0.884	0.878	0.888	0.911	0.952	1.013	1.080	0.912
		1.00	0.976	0.955	0.938	0.927	0.924	0.928	0.946	0.986	1.057	0.872
2	Cruiser.....	1.00	0.979	0.962	0.959	0.963	0.972	0.978	0.993	1.015	1.042	0.806
		1.00	0.981	0.964	0.955	0.961	0.983	1.009	1.036	1.067	1.109	0.822
3	Destroyer.....	1.00	0.977	0.955	0.938	0.928	0.931	0.947	0.975	1.020	1.074	0.784
		1.00	0.980	0.965	0.959	0.959	0.964	0.975	0.993	1.017	1.050	0.751
4	Tug.....	1.00	0.987	0.950	0.947	0.948	0.956	0.970	0.992	1.024	1.067	0.980
		1.00	0.995	0.966	0.933	0.929	0.942	0.967	0.999	1.036	1.086	0.944
5	Cruiser.....	1.00	0.978	0.964	0.969	0.980	0.986	0.989	0.995	1.011	1.042	0.800
		1.00	0.979	0.957	0.943	0.946	0.964	0.984	1.010	1.037	1.062	0.806
6	Cargo vessel...	1.00	1.010	0.994	0.958	0.946	0.953	0.972	0.994	1.021	1.052	0.967
		1.00	0.993	0.978	0.976	0.991	1.004	1.018	1.031	1.044	1.058	0.895
7	Whaler.....	1.00	0.996	0.928	0.930	0.944	0.961	0.980	1.000	1.022	1.053	1.049
		1.00	0.992	0.973	0.968	0.970	0.981	1.003	1.036	1.079	1.134	0.970
8	Tanker.....	1.00	1.001	0.964	0.966	0.970	0.974	0.981	0.991	1.004	1.020	1.013
		1.00	0.996	0.981	0.966	0.965	0.972	0.985	1.003	1.031	1.080	0.985
9	Channel steamship...	1.00	0.979	0.951	0.939	0.946	0.965	0.988	1.014	1.047	1.086	0.921
		1.00	0.970	0.961	0.970	0.994	1.017	1.040	1.062	1.084	1.107	0.852
10	Whaler.....	1.00	0.997	0.913	0.904	0.907	0.920	0.945	0.982	1.021	1.061	0.968
		1.00	0.996	0.973	0.946	0.932	0.937	0.955	0.978	1.013	1.063	0.909
11	Cargo vessel...	1.00	1.008	0.989	0.946	0.936	0.944	0.963	0.988	1.017	1.047	0.930
		1.00	1.010	0.997	0.970	0.958	0.966	0.987	1.007	1.025	1.044	0.886
12	Trawler.....	1.00	0.987	0.939	0.925	0.924	0.934	0.955	0.979	1.003	1.028	0.931
		1.00	0.968	0.951	0.948	0.952	0.961	0.973	0.989	1.016	1.062	0.876
13	Depot.....	1.00	1.005	0.988	0.955	0.949	0.959	0.975	0.994	1.021	1.058	0.940
		1.00	0.983	0.974	0.982	1.012	1.037	1.056	1.073	1.089	1.104	0.828
14	Channel steamship...	1.00	0.968	0.940	0.917	0.916	0.947	0.984	1.021	1.066	1.132	0.978
		1.00	0.948	0.921	0.914	0.943	0.988	1.036	1.081	1.123	1.167	0.913
15	Destroyer.....	1.00	0.961	0.948	0.959	0.971	0.987	1.004	1.028	1.060	1.112	0.944
		1.00	0.976	0.948	0.943	0.958	0.988	1.021	1.054	1.089	1.123	0.916
16	Cargo vessel...	1.00	1.014	1.001	0.976	0.969	0.974	0.995	1.027	1.062	1.099	0.949
		1.00	1.005	0.988	0.968	0.965	0.977	0.999	1.025	1.051	1.080	0.935

Table I shows how the B R values at every 10-degree angle of heel up to 60 degrees can be obtained for the prism, and the calculation proceeds similarly up to 90 degrees. The values of B R at the various angles of heel are found to be practically the same as those obtained by use of the integrator.

The breadth (x/2) in way of the deck edge, when it occurs as in Figure 2, should be reduced so as to give the correct area of that particular wedge.

Calculations have been made for a number of forms, representing actual vessels in loaded and light conditions. Some of the results are shown in Table II, together with ship/prism ratios or coefficients.

The ship/prism ratios in Table II are not the actual ratios, but the corrected or "basic" ship/prism ratios. This correction was made because it was considered that basic ship/prism ratios are more convenient, and is arrived at as follows:—

The actual ship/prism B R ratios were tabulated and curves drawn. It was observed that, while all the curves were more or less of the same characteristic shape, the B R ratios at 1 degree angle of heel showed wide variation, depending on the fineness of the water-lines (i.e., the initial B M). It was decided that a better comparison of the data could be obtained by making the ship/prism B R ratio equal to unity at 1 degree by adding the requisite amount. At any other angle this 1-degree correction is modified according to a straight-line law, until, at 90 degrees, the basic ratio is equal to the actual ratio.

The example in Table III illustrates the method of application.

The B R ratio at 90 degrees may be approximated by the following formula:—

Let

$$x = \frac{\text{Topside sectional area of prism}}{\text{Total sectional area of prism}},$$

$$y = \frac{\text{Topside coefficient of fineness of ship}}{\text{Displacement prismatic coefficient of ship}};$$

then ship/prism B R ratio at 90 degrees

$$= \frac{x\,(y-1)}{(x+1)} + 1.$$

An approximation to the B M for the prism may be obtained from the formula:—

$$\text{B M of prism} = B^3/12A,$$

where

B = breadth at water-line, and

A = immersed sectional-area of prism.

The estimate for the B M of the ship is dependent on the moment of inertia of the water-plane and the volume of displacement. The moment of inertia may be approximated by the following formula:—

$$I = \left[\frac{B^3 \times L \times (3\,C - 1)}{24}\right] Y,$$

where

I = transverse moment of inertia of water-plane,

C = water-line coefficient of fineness, and

Y = inertia coefficient, as given in Table IV.

TABLE III.—APPLICATION TO A TRAWLER OF SIMILAR LINES TO NO. 12 (TABLE II). LOADED CONDITION.
Ship/Prism B. M. Ratio = 0.910.

Angle of heel, in degrees.....	1	10	20	30	40
Basic B. R. ratios (Table II)..	1.000	0.987	0.939	0.925	0.924
Correction due to B M ratio*..	−0.090	−0.080	−0.070	−0.060	−0.050
Ship/prism ratios (R).....	0.910	0.907	0.869	0.865	0.874
Prism B R (calculated) (P)....	0.086	0.862	1.757	2.358	2.673
Ship B R (P × R).....	0.078	0.782	1.527	2.040	2.337
Angle of heel, in degrees.....	50	60	70	80	90
Basic B. R. ratios (Table II)...	0.934	0.955	0.979	1.003	1.028
Correction due to B M ratio*.	−0.040	−0.030	−0.020	−0.010	—
Ship/prism ratios (R).....	0.894	0.925	0.959	0.993	1.028
Prism B R (calculated) (P)...	2.778	2.718	2.529	2.243	1.887
Ship B R (P × R).....	2.483	2.515	2.425	2.227	1.941

* The correction adopted at 1 degree (—0.090) is that necessary to bring the B R ratio from 1.000, the basic figure, to 0.910, the actual B M ratio in this example. The corrections at other angles are modified lineally so as to be zero at 90 degrees.

TABLE IV.—INERTIA COEFFICIENTS, Y.

Water-line Coefficient, C.	Transverse Inertia Coefficient, Y.
0.65	0.925
0.70	0.935
0.75	0.945
0.80	0.955
0.85	0.965
0.90	0.975

TABLE V.—COEFFICIENT, c.

Type of Vessel.	Value of c	
	Loaded.	Light.
Full cargo steamship.....	1.005	1.035
Large twin-screw oil-tanker.....	0.981	0.980
Cruiser.....	1.003	0.982
Destroyer.....	1.015	1.020
Trawler.....	0.992	0.989
Large passenger liner.....	0.995	0.995
Cross-channel steamship.....	0.991	0.992

A similar formula often used for full cargo vessels is:—

$$B\,M = \frac{B^2 \times (3\,C - 1)}{25 \times d \times \text{block coefficient}}.$$

where d = moulded draught of the vessel.

B R at 1-degree is given by B M × 0.01745.

In order to draw the stability curve of the proposed vessel, it is necessary to know the position of the center of buoyancy and of the center of gravity in the condition for which the curve is required. The righting levers G Z can be obtained by means of Attwood's formula:—$G\,Z = B\,R - B\,G \sin \theta$, B G being the vertical distance between center of buoyancy and center of gravity in the upright condition.

The vertical position of the center of buoyancy above the base-line can be approximated from the formula:—

$$K\,B = \left[d - \frac{1}{3}\left(\frac{d}{2} + \frac{V}{A}\right)\right] c,$$

where

V = volume of displacement,

A = area of water-plane, and

c = a coefficient having the values given in Table V.

In preliminary design work, the increment in B R due to the buoyancy of deck erections at large angles may be estimated approximately by taking the depth of the prism to be equal to the moulded depth of the ship plus the height of erection plus one-third of the mean sheer. The increment thus obtained would be multiplied by the fraction (E/L), where E = length of erection and L = length of ship.

MISCELLANEA OF INTEREST TO NAVAL ENGINEERS.

HOW CAN WE DEVELOP INVENTORS?—This paper by Charles F. Kettering was contributed by the Committee on Education and Training for the Industries and presented at a recent Annual Meeting of the Ameircan Society of Mechanical Engineers. It is reprinted from Mechanical Engineering, April, 1944. Mr. Kettering is Vice-President and Director of Research of the General Motors Corporation.

I think this question of how can we develop inventors, or inventions, because that is the real problem, is one that should concern us greatly. There are many different ideas of what an invention is. The inventor has one idea, the manufacturer has a different one, and the patent attorney has still another. Sometimes it takes quite a high order of intelligence to detect any similarity between them. I mean this seriously—it is a common thing for an inventor not to understand the patent on his own invention.

Likewise, there is no sharp line of demarcation between invention and research except for this distinction—that invention has to do with a specific result, while research is concerned with the determination of those factors which may be necessary in the development of that result.

Some years ago a survey was made in which it was shown that if a person had an engineering or scientific education, the probability of his making an invention was only about half as great as if he did not have that specialized training.

Now that is very interesting, and I have spent a great deal of time wondering why it is so. As a result, I have arrived at a definition of what an inventor is. An inventor is simply a fellow who doesn't take his education too seriously.

You see, from the time a boy is six years old until he graduates from college he has to take three or four examinations a year. If he flunks once, he is out.

But an inventor is almost always failing. He tries and fails maybe a thousand times. If he succeeds once, he is in. These two things are diametrically opposite.

We often say that the biggest job we have is to teach a newly hired boy how to fail intelligently. We have to train him to experiment over and over and to keep on trying and failing until he finally learns what will work.

We also have to teach him that everything is not in the books. In his education he invariably gets the idea that this is so because his textbook is always the last word and final authority on whatever he is studying. If we fail to do this, sooner or later he will say, "There is no sense trying this experiment because page 284 of this book says it won't work."

Then we have to explain that these things we are doing have never been done before. If they had, we wouldn't go to all the trouble of repeating them. The books used to say we couldn't build a Diesel injector to operate at 20,000 pounds per square inch. What it really meant is that an injector couldn't operate at that pressure if it was built as the book specified. Today we have hundreds of thousands of injectors operating at pressures much higher than that. The point is simply that no matter what we are doing, if it is new we can always find a book that will tell us it can't be done. Any time we want to stop a project, all we need to do is assemble a committee and tell them about it. Immediately they will tell us we were crazy to start it. So to protect ourselves we have made this very simple rule: When you start a new problem, stay away from the library because it may handicap your thinking.

In training an inventor, how closely do we want him to work with the library? Should he see what has been done first, or should he start out with his own ideas and then go back to the books after he has got a little way along?

The first step is always to set up the problem. When we have done that, we say, "Here's the way we think we ought to proceed," and then go to the library to see what others before us have thought about it. But if we go to the library first, few of us are strong enough to stay out of the rut that may be set up. So another thing we must do in developing an inventor is to teach him to analyze his problem. If he decides that it is worth working on at all, what is the probable first experiment he should try?

Many people want to go beyond the first experiment in their predictions. "If this experiment works," they say, "we will do such and such next." But we should not worry about that. If we knew what was going to happen, we wouldn't need to experiment in the first place. We should realize that prior to the first experiment our ignorance factor may be as much as 100 per cent.

"But suppose the experiment doesn't work the way we think it should?" they ask us.

The chances are ten to one it won't. That shouldn't deter us, however, from going ahead and trying it. After all, an experiment that fails is simply an experiment in which we are wrong. Rarely does an experiment mislead us, providing we are intelligent enough to follow as it directs.

If an experiment doesn't work the first time, it must be tried again; and when a successful experiment is obtained, a continuity can be laid out for going on to the next step. We can then decide what the second experiment should be and go through with it in exactly the same manner. Since none of us is ever smart enough to arrive directly at the final result, we must work our way very laboriously from experiment to experiment and from test to test until we finally get there.

A lot of people don't want to do this. They would like to find a short cut for the tedious trying and failing of experimentation. They think they are smart enough or educated enough to get the result directly. Unfortunately, the only people who have ever accomplished anything in science or engineering are the ones who never thought they were smart enough to get irked by having to try things to see if they would work.

A man who had seen a test of one of our Diesel engines came to see me the other day. "That is a very efficient engine," he said. "I would like to talk to your thermodynamics expert about it."

"I am sorry," I replied, "we don't have anyone here who even understands the word 'thermodynamics,' much less be an expert on it. But if you want to know how we developed this engine, I'll be glad to show you."

I took him to our dynamometer room and showed him our single-cylinder setup and told him how we had tried one thing after another for about six years until the engine itself finally told us exactly what it wanted.

"Goodness," he said, "that's an awfully tedious way to design an engine."

"It is," I agreed, "but it is the only way we know."

We don't know how to do any new job without going through all that tedious cut-and-try. Therefore, the next thing we must tell these young engineers is that tediousness is something they must endure and not resent even though they have a technical education.

These things seem extremely simple, but they are basic to the problem of developing inventors.

The point is that we don't solve a technical problem or make an invention in the laboratory. We do these things in our heads. All this laboratory apparatus is simply to help us get an understanding of the problem we are working on. I think the thickness and density of the human skull can be very well represented by the amount of apparatus we sometimes need simply to force an idea through a quarter inch of skull bone. That is why it is so important that we develop proper attitudes in the people we are training to be inventors.

About ten years ago we started a new study of the strength of materials which was based entirely on testing a production piece under simulated working conditions. In other words, if we were interested in a crankshaft, we would build a testing machine that would load that crankshaft exactly as it is loaded in the engine, that is, a load fluctuating so fast with so much tension, so much torsion, at such a temperature, etc. We did that because we had learned that tests on standard test specimens gave us information of a very unsatisfactory kind and that when we tried to test a piece to failure in an actual engine the resulting debris was such that we couldn't tell just what had happened.

In our machine we put anticipators on a piece so that the instant a failure started the load would be released automatically and we could see exactly where the trouble had started.

With this machine we have been able to improve the fatigue life of many mechanisms as much as 500 per cent simply by seeing *how* and *why* the failure starts and by then making very small changes in fillets or slight shifts in temperature during heat-treatment.

We couldn't have calculated these things. No one is smart enough to make such calculations. It has to be a very primitive sort of study which goes back to the very elementary process of trying a thing, learning its weaknesses, and then trying something else.

One of the problems which a research laboratory will always have is how to get along with the manufacturing end of the particular organization to which it belongs. The production man has a problem, and he brings it to us to solve. We analyze it and write a report on our solution. If we use a lot of logarithmic curves and fancy nomenclature the production man never heard of before, he gets mad because he can't see that we have helped him very much. I don't blame him.

To avoid that difficulty, we have worked out a very simple procedure. Before a special report is distributed, it is carefully read by someone in our organization who knows the person for whom the report is intended, and a little circle, or goose egg, is drawn around every word that he thinks may be misunderstood. Then we call in the research engineer and ask him to read the report aloud—but skipping the words that are circled. He starts reading and then exclaims, "My goodness, if you are going to leave out all those words, it won't make any sense at all!"

"All right," we say, "perhaps the reader won't understand those words, so it is just the same to him as if they weren't there at all. If we are going to help with this problem, we must use words that will be understood."

One of these reports had to do with noise and the word "decibel" appeared several times on every page. The boys couldn't find a simpler word for decibel, so with every copy of that report we supplied a little noise box that had ten keys on it. If the report said the noise level was seven decibels, the reader just punched the key marked "seven" and he could hear exactly what seven decibels of noise was.

This question of words and the meaning that we attach to words has always been very interesting to me. It is like the formula problem in which, if a fellow knows the formula for something, he thinks he understands all about it. A lot of people think that if they know the name of something, that's the whole story.

As an illustration, I once asked a famous scientist this question: "Why can I see through a pane of glass?"

"That's very simple," he said. "Glass is transparent."

"I'm afraid the word 'transparent' means nothing at all to me," I replied.

"The word 'transparent' simply means anything you can see through," he said. "Why do you want to know that?'

"I would like to know *why* I can see through a pane of glass. I would like to know whether light waves travel through the glass as light—or whether they are received and rebroadcast in some other form from molecule to molecule. If I knew that, I would be better able to decide some other things I would like to know."

"The trouble with you," he said, "is that you are a modelist; you want an image of something. Modern science doesn't form images any more. We stopped that a hundred years ago. Give me a better problem—something a little more practical"

"All right, here's one: When I rub my hands together, why do they get warm? Why shouldn't they get cold?"

"Oh, that is quite simple—it is due to friction."

"Yes," I said, "now what is friction?"

So we argued back and forth for three quarters of an hour and finally we came to the profound conclusion that we don't know anything about friction—except that friction is a thing that makes your hands warm when you rub them together—and we don't even know how it does that.

A few yeras ago I assigned a young man to work on this friction problem and had him rub little pieces of metal to see how much they wore away, how much heat they generated. and so on. After he had been working on it for a little while, I stopped in to see him. "How are you coming along?" I asked.

"Not very well," he replied. "Don't you think this is an awfully childish problem for a fellow who has the kind of education I have?"

I didn't think so, or I wouldn't have assigned him to it, so I asked what the trouble was.

"It wouldn't be so bad," he exclaimed, "if it wasn't all in the books!"

"I'm glad you mentioned that," I said, "because if we have you working on this thing when it's all in the books, like you say, then we ought to stop right away. But before we do that, I want you to show me that book. If you can find that book or any book like it, I will give you a medal."

I saw him about a month later and asked how he was progressing on the medal business.

"Awful," he said, "I haven't been able to find anything. And I can't understand that, because it's something so fundamental I should think a lot would be known about it. I can't understand it at all. There's not a thing in the books about it."

I want to tell you something of the outcome of this episode because it shows how much we can accomplish if we don't place ourselves too high above our jobs. This same young fellow soon discovered some very interesting fundamental facts about friction and wearing surfaces which have enabled us to produce better piston rings for Diesel locomotives than we ever had before. When we first started to run our locomotives, we had to change rings every 50,000 miles. No one objected to that. They thought 50,000 miles was about

as long as a piston ring ought to last. Now we have rings that will last 500,000 miles, and I think we ought to be able to get them to last 1,000,000 miles before we're done.

It was in connection with these same engines that someone once asked how it happened that we were making most of the Diesel-electric locomotives in the country. "You must have awfully good patent protection," he said.

"Well, here's the reason," I said. "You see, a great many people think we're crazy. That is much better protection than any patent."

I would rather have my competitor think I am crazy than have a stack of patents a mile high, because he can easily get around my patents, but if he thinks I am crazy, he won't even bother to check up on them. I was in the automobile starting, lighting, and ignition business for five years without any patent protection at all to speak of, because everyone knew that the breaker mechanism in my ignition system was crazy, and that it wasn't worth copying. They infringed every patent I had, but they didn't infringe my ignorance factor.

I often tell the boys this story: I have been making inventions and taking out patents for many years. When I first began to apply for patents, most of my inventions had been patented 50 or 60 years before. Later I was only 40 years behind—then 30 years—and so on. So I drew a set of coordinates and plotted just how far behind I was on each invention. There was a gentle slope to the curve, so I could extend it to the base line to see how old I would be when I made an original invention. It comes to about 125 years.

If we can succeed in eliminating from a young man's mind the idea that just because he thought of a thing it must be very good, then we can go ahead without the danger of temperament getting into engineering.

We have a little rule in our laboratory which we think solves this problem. It's a very simple rule and the only rule we have. It is just this: "The Job Is the Boss." What does the engineer think of this new piston? That doesn't matter. What does the engine think about it? That matters. The engineer's opinion is worth very little. The engine's opinion is worth a great deal.

If the engine says, "I like this piston," and it happens to be contrary to the engineer's pet idea, that's too bad. It simply proves that the engineer was wrong. After all, as we said, the only reason for all this expensive research is that it corrects our ignorance factor so that we can see the problem in its true light.

Opinions are not the only things that lead us astray. Sometimes we are led into difficulties by certain aspects of the so-called practical side of our educations. We accumulate rules of thumb, arbitrary constants, and formulas. We accept these as fundamental. We should question them whenever we encounter them. They are the stumbling blocks of progress.

There is a formula by which we can calculate the fatigue life of a piece of spring steel which is being stressed under certain conditions. This formula tells us how many thousand cycles of stress the spring will undergo if its surface is perfectly mirror-smooth, how many less if it is slightly rough, and how many still less as its roughness increases. We take a flat piece of smooth spring steel about 3 inches wide and 2 feet long and put it in a test machine and bend it back and forth until it breaks. We find that the standard spring, as furnished by eight or ten different steelmakers, will break in about the same time—at a couple thousand cycles. That checks very closely with the formula. We ask each of the steelmakers to run the same test and their results check with ours. The formula and the experiment agree. Everyone agrees that it is a wonderful proof of the formula and shows there is no use trying to run a spring any longer than the formula says.

Then we say, "Suppose we bang up this piece of spring steel until it is rough all over—then how long will it last?"

After thinking about this for a while, they say, "We don't know how many cycles of stress it will stand, but we all agree that its fatigue durability will be seriously impaired."

So we asked them to make us some more samples. "Put a little mark on them so you can identify them as your pieces," we say. Then we give these

samples a little treatment which we call shotblasting, or surface peening, in which we shoot a lot of little hard steel balls at each sample until it is thoroughly roughened. That done, we send the samples back to the steelmakers to be tested.

Then a surprising thing happens. Instead of breaking at 2000 cycles, the samples are flexed up to 2,000,000 cycles and still they don't break. The formula still says they will break at 2000, but they don't know that. They keep right on flexing at 2,000,000. That is quite a gain in per cent.

We have learned that in most problems there is a normal time rate of development which corresponds very closely to the normal yearly increment of improvement made by industry. There is not a great deal that we can do to speed up such improvements, because they are so intimately geared into the whole system that they can be assimilated only at a certain rate. But in other problems, such as certain medical problems, the question of how long it is going to take to solve them is of relatively little importance. The results will justify any expenditure of time. All that is necessary is that the man who is going to work on the problem be shown that it is worth solving, or he won't want to try it.

We had some experience with this in the early days of the ethyl-gasoline development. We had been working on the so-called aromatics for a number of years and had some pretty good results, but nothing really exciting. So some of the boys came up to my office and said, "We are young yet, and we would like to do something in the world, and we don't want to be kept on this problem forever, because we don't see any hope of ever working it out."

I was leaving for New York that afternoon, so I said, "Let me have a couple of days to think this over, and when I come back I'll see if I can't work out something for you."

I thought that it would be wrong to stop this work, even though they had been correct when they said our progress to date was very discouraging. When I got on the train to go back to Dayton, I noticed a newspaper lying on the seat beside me and I picked it up. On the first page there was a little item entitled, "University Professor Discovers Universal Solvent."

That interested me because I happened to know an amusing story about two chemists who were trying to discover a universal solvent. It seemed they talked a banker into loaning them some money and set up a little laboratory just outside of town. One winter morning a farmer had a blowout outside their place, so he came in and asked if he could use their telephone to call up the garage. They said, "Sure, go right ahead." When he was through telephoning, he asked if he could wait there until the garage brought him a spare tire and they said, "Sure, you can." So he stood there watching them for a while, and finally he asked what they were doing.

"We are working on the most important thing in the world," they said. "We are developing a universal solvent."

"What does that mean?" asked the farmer.

"Well, that is a liquid that will dissolve anything you put into it."

"My goodness," said the farmer, "that is marvelous, but what in tarnation are you going to keep *it* in?"

Because I happened to know that story, I was interested in this so-called universal solvent and I read the newspaper story in its entirety. The material referred to was selenium oxychloride.

The next morning when I got to the laboratory I showed this newspaper item to the boys and said, "Here's one we'd better try before we quit," so they went ahead and got some. It turned out to be almost a universal solvent in its action on valves and rings, but it did something that none of us had anticipated. Theoretically, selenium oxychloride should have made the knock worse because it contained both chloride and oxygen, which had been knock-inducers in every other compound we had tested. But instead of inducing knock, selenium oxychloride turned out to be a powerful knock-suppressor. That was our first step into the study of the metallo-organic compounds, which led us through a great many experiments until we finally arrived at tetraethyl lead.

The point of this story is that we must always know whether a problem is worth solving. There will always be dark days, but if your conviction of the value of the problem is such that you go right ahead in spite of the difficulties, the chances are that you will achieve success in the end.

I like to think that we can take even a highly educated young man and teach him to be an inventor without ruining his education. I have only one complaint against our educators and it is that they haven't always told us the complete story.

A friend of mine who teaches general medicine at one of our large universities once told me this story about education. He is an old hand at teaching medicine. He lectures to the classes, teaching the boys to be general practitioners. His system is sound and doesn't differ radically from similar courses in other universities, except for the last lecture he gives the group. That lecture is unique. On this occasion, he says, in part:

"Young men, we are together for the last time. We have had a very pleasant time. You have been a good class and I have enjoyed working with you.

"I have given you the best information available—the best case histories I could find. The textbooks we have used are the most widely accepted and reliable. But before we part company, I want to caution you that the science of medicine is developing so rapidly that in a few years from now perhaps half of the things I have taught you won't be so. Unfortunately, I don't know what half that will be."

I have been very interested in co-operative education because I feel that by bringing a boy in contact with both the school and industry we can lap-weld him to a job instead of butt-weld him. I worked with Dean Schneider at Cincinnati for a number of years. I have also been interested in a co-operative system at Antioch; and right before Dean Schneider died, I helped to get him to go out to Northwestern University to set up the co-operative system there.

My interest in co-operative education is not limited to having the boys go to school half time and work in industry half time. I don't care how it is done, but I would like to see the university and industry work closer together because I think they should understand each other's problems. I think if we could do that, it would be much less necessary to teach trade subjects in school. We could spend more time on the fundamental broad principles of physics, chemistry, and mechanics.

There are several other ways that this result can be achieved. One is by alternating the professors instead of the students. The instructor could teach half time and, as I once said, work half time. Somebody objected to my saying that, so I apologized, but you know what I mean.

I don't care how we achieve this co-operation, whether it is half-timing the boys or half-timing the instructors, or doing research, just so the educational institutions can teach these boys the fundamentals they will need in their work, and industry can teach them the practical applications it will need from them in its work.

If we can just achieve the broad outlook and be honest with ourselves and admit that the things we know are so small compared with the unknown, I think we will see that there is no end to progress. I am not worried about the libraries we have today—I am worried about those infinitely greater libraries that haven't any books in them yet. Every year is going to add new things and new books that we don't know. I am interested in getting some of those books written.

We should date-line every bit of information we have. We should say, "This is what we know as of today." We shouldn't worry about what it was yesterday—or what it is going to be tomorrow. We should recognize that ideas change and grow, and we should gear ourselves to the most progressive thinking. If we can do that successfully, we will have no difficulty in teaching men to become inventors, because inventing is simply a state of open-mindedness.

SOLAR SEARCHLIGHT.—Members of this Society should have a peculiar interest in this report of a signaling mirror for lifeboat use which accurately directs high intensity sunbeams towards potential rescuers. The designer, Wentworth M. Potter, is also author of this article reprinted from General Electric Review, May, 1944. Mr. Potter is employed in the Engineering Division, Lamp Department, of the General Electric Company.

In the extended air, naval, and land operations of World War II, the possibility that men will be adrift at sea in life-jackets or rafts, or lost in isolated country, has unhappily often been realized. With emergency supplies, men may survive a considerable period of such severe hazards; but if lives are to be saved, early rescue is usually imperative. Hence, survivors are being provided with compelling means for attracting the attention of potential rescuers.

An attention-compelling signal is no less needed by day than are lighted beacons by night. To an aircraft pilot, with a vast expanse of sea in view, a life-raft is an easily overlooked object against the broken pattern of water; a single man is still less discernible.

Many have related the agony of seeing, but not being seen by, ships and aircraft. For example, in *Life* (Nov. 2, 1942) John Hersey related the Coral Sea experiences of Lieutenants A. W. Anderson and J. P. Van Hour and their crew:

> First Day: "About 11:30 (A.M.) we spotted a Navy PBY coming over close. We fired four flares but received no response . . . "
>
> Second Day: "Another plane flew up, seemingly straight toward us. It passed not more than a mile away. God, we could practically see the pilot's eyes. But it went on by."
>
> Seventh Day: (two had died . . .) ". . . around two o'clock a Navy scout plane came over close. We all waved shirts and flashed signals with can tops and spread so much aluminum slick . . . He soon came back and landed. . . ."

All have noticed how effectively attention is attracted by flashes of sunlight reflected from windshields of distant cars. Why not, then, utilize an efficient mirror to direct a powerful beam of sunlight to ship or airplane? A shiny can top is a lot better than nothing; but when life is at stake, a person wants as powerful a flash as possible, and, above all, assurance of hitting his target.

The need for a scientifically-designed solar-signaling mirror which could be accurately aimed was voiced by the United States Coast Guard through the National Inventors' Council. Such a mirror, of high efficiency and simple construction, is now being manufactured in large quantities for the Armed and Maritime Services by the General Electric Company. It was designed by the writer in co-operation with the National Bureau of Standards at Washington (D. C.).

The device was developed to combine features which appeared to be primary requirements for an effective emergency signaling mirror:

(1). A flat surface of high reflecting power which will stay bright under severe salt air or water exposure;

(2). Resistance to impact. abrasion, and bending, which would rapidly reduce beam intensity;

(3). Convenience in use and insurance against loss;

(4). Integral means of precision aiming.

In addition, it was designed for efficient large-scale fabrication with existing facilities, which were developed initially for automobile lamps.

The mirror, as indicated in Figures 1 and 2 (and in the COVER illustration), consists of a small rectangle of tempered (heat-treated), polished plate glass with smooth chamfered edges and rounded corners. It may be securely suspended from the neck by means of the braided lanyard looped through a corner hole. The rear surface of the glass carries a thin film of vaporized aluminum of high reflectivity, which forms the main mirror (Figure 1), and

the round rear mirrir seen in Figure 2. A central aperture in the form of a cross is left clear. All edges and the rear side, except for the round mirror, are coated with black lacquer, on which are printed illustrated instructions for use. Then the edges and entire rear side are sealed with tough, baked, clear lacquer.

Efficient Reflection Produces Powerful Beam of Sunlight.

These aluminized mirrors reflect a large proportion of the incident sunlight, having a reflection factor between 0.80 and 0.85, or some 30 per cent higher than that of chromium plate. Thus, they are as effective as new chromium mirrors of substantially larger size. Under most favorable sunlight conditions, the beam candlepower appears to be of the order of 15 to 16 millions.

The degree of planarity or flatness achieved in tempered plate glass is excellent for the purpose. The distortions incident to the tempering process are kept within limits which widen the beam only moderately. The small reduction in candlepower is compensated by greater steadiness of signal. Since glass is rigid, the effectiveness of the reflected beam cannot be destroyed by bending or distortion in use.

The lacquer coating precludes any substantial loss in service of either effective mirror area or reflection factor. It is not only resistant to salt air and water but is tough and resistant to ordinary abrasion. As additional insurance against years of stand-by in lifeboats or under other adverse conditions before being needed, many of the mirrors are supplied in moisture-proof packaging.

Strength.

The use of lacquer protection for the mirror surface permitted the use of a single plate of glass of a thickness favorable for developing strength through tempering. Though no known glass is absolutely breakage-proof, tempering imparts very much greater resistance to fracture from impact. Of at least equal importance, particularly in inflated rubber life rafts, is the fact that if the surface of tempered glass is broken, by a pointed instrument, for example, it does not form sharp dangreous splinters; instead, it disintegrates into small, blunt crystals.

Convenience.

The dimensions of the mirror are only four by five inches; it is easily held while in use or carried in a pocket when not in use; its weight is less than eight ounces. The braided cotton lanyard keeps it at hand in an emergency and prevents loss.

Precision Aiming.

A beam of sunlight reflected by a plane mirror is narrow; its spread is only about one-half degree, or, in diameter, about 50 feet at one mile. It is easy to direct such a beam to a nearby target by noting the position of the spot produced by the beam. But when the target is at some distance, perhaps on the horizon or in the air, the spot can no longer be seen, and a means of precision aiming is required. It is obviously desirable that facilities for aiming be integral with the mirror and that they do not invariably require the use of both hands.

In the General Electric Signaling Mirror, the means for aiming consist only of the central cross-shaped aperture; and the round section of aluminum film left visible on the rear side (Figure 2). Thus, they add neither bulk nor weight, and no additional material is employed. As first suggested by the writer, the central aperture was in the form of a small round hole, which fulfills all requirements as to precision of aim. However, the cross form suggested by his collaborators was desirable for this service as it facilitated finding and following the target, and sunlight shining through it produced a spot of distinctive shape, thus facilitating instructions for use.

FIGURE 1.—HERE THE AUTHOR SHOWS THE MIRROR AND HAND SO POSITIONED AS TO ILLUSTRATE ONE OF THE MANY POSSIBLE RELATIONSHIPS OF MIRROR, MIRROR SHADOW, CROSS-SHAPED SPOT OF SUNLIGHT, AND LINE OF SIGHT TO TARGET. SPOT WILL OFTEN BE FOUND UPON FACE, SHIRT, OR OTHER SURFACE. IT IS SOMETIMES MORE CONVENIENT TO HOLD THE MIRROR CUPPED IN ONE OR BOTH HANDS. THIS IS PARTICULARLY TRUE WHEN THERE IS A LARGE ANGLE BETWEEN SUN AND TARGET; ONE HAND, AT END OF MIRROR FARTHEST FROM FACE, MAY AT THE SAME TIME HOLD MIRROR AND INTERCEPT THE SUNLIGHT COMING THROUGH THE APERTURE.

FIGURE 2.—HERE IS ILLUSTRATED RECOGNITION OF THE CROSS-SHAPED SPOT OF SUNLIGHT AND ITS VIRTUAL IMAGE IN THE ROUND REAR MIRROR.

FIGURE 3.—A SCHEMATIC ILLUSTRATION OF THE AIMING PROCEDURE, WHICH IS PRINTED ON THE OUTSIDE OF THE PACKAGES WITH INSTRUCTION TEXT SIMILAR TO THAT IN FIGURE 2. THE REFLECTION OF THE HAND AND SUNLIGHT CROSS WOULD APPEAR TO THE USER AS SHOWN IN THE COVER ILLUSTRATION.

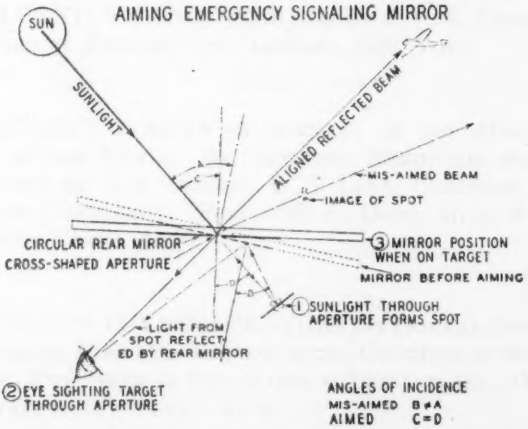

FIGURE 4.—THIS PRINCIPLE OF AIMING OBVIOUSLY ALLOWS LATITUDE IN LOCATION OF BOTH EYE AND SPOT IN RELATION TO THE MIRROR, THOUGH THE APERTURE USED IN THIS MIRROR WAS DIMENSIONED ON THE ASSUMPTION THAT IT WOULD MOST OFTEN BE EMPLOYED AT ONLY A FEW INCHES FROM THE EYE.

Note that sun, aperture, and cross-shaped spot are always in alignment; when the beam is on the target, these, the user's eye, the virtual image of the spot, and the target all lie in the same plane.

As indicated in the abbreviated instructions printed on the back of the mirror and by the sketch in Figure 3, the procedure of aiming the reflected beam is a simple one. The user positions the mirror to face a point about halfway between sun and target, picks up the image of the sunlit cross in the round, rear mirror, and sights target through center of cross. The conditions for accurate aiming (Figure 4) are completed when the mirror is tilted so as to superimpose the image of the spot on the aperture and target.

The inherently parallel opposite surfaces of the very thin aluminum film best insure coincidence of beam with line of sight. Obviously, if they were out of parallel by only a small fraction of a degree, the narrow reflected beam might miss the target entirely.

The film's microscopic thickness is also an advantage since it permits viewing and centering targets at angles as high as 80 degrees from the perpendicular to the mirror surface. Were reflecting material of substantial thickness used, the deeper aperture would act as a louver, restricting the angle of view. Compensatory measures, such as enlargement of hole or countersinking of front surface, would, respectively, make the centering of target more difficult and involve loss of effective mirror area.

There are, of course, two conditions under which the aiming method is not applicable: (a) when target is silhouetted directly against a bright sun, and (b) when it is 180 degrees away from the sun. In locations near the latter point, the user's head may place the mirror in shadow or the mirror may be so greatly foreshortened as to be ineffective. If the angle is such that aiming cannot be accomplished with either eye, a second specular surface, if available, can be used to direct sunlight to the signaling mirror at a more favorable angle.

That the mirror is practical even under extreme conditions has been conclusively proved in official tests conducted by government authorities. Results show that a man in a life jacket and in rough water can consistently signal distant ships, aircraft, or other vantage points.

BOOK NOTICES.

Note:—Because of difficulties attending the publication of the Journal, it will be impossible to include notices of new books which are received later than the tenth of the month preceding publication.

QUESTIONS AND ANSWERS FOR MARINE ENGINEERS FROM MARINE ENGINEERING AND SHIPPING REVIEW, AUXILIARIES—APPLICATION OF STEAM AND HEAT IN PRODUCING POWER—POWERING, FUEL ECONOMY, PROPULSION, PROPELLORS AND SHAFTING. COMPILED BY CAPTAIN H. C. DINGER, U. S. N. (RETIRED). PUBLISHED BY SIMMONS—BOARDMAN PUBLISHING COMPANY, 30 CHURCH STREET, NEW YORK, N. Y. $2.00.

20TH CENTURY ENGINEERING BY C. H. S. TUPHOLME. PUBLISHED BY PHILOSOPHICAL LIBRARY, NEW YORK.

MERCHANT FLEETS—A SUMMARY OF THE MERCHANT NAVIES OF THE WORLD. BY CRITCHELL REMINGTON WITH INTRODUCTION BY VICE ADMIRAL E. S. LAND, CHAIRMAN U. S. MARITIME COMMISSION. PUBLISHED BY DODD, MEAD & COMPANY, NEW YORK. $4.00.

PHYSICS OF THE 20TH CENTURY BY PASCUAL JORDAN—A SYSTEMATIC COMPLETE REVIEW OF THE CONCEPTS OF MODERN PHYSICS. PUBLISHED BY PHILOSOPHICAL LIBRARY, INC., 15 EAST 40TH STREET, NEW YORK. $4.00.

ASSOCIATION NOTES.

The War Department has authorized the publication of the following list of Inventive Problems, solutions of which are of present interest to armed services:

1. A relatively simple gage to measure the impulse of explosion blast, positive and negative phases to be determined separately but concurrently. It would be desirable if the duration of each phase could be determined in some simple manner.

2. A waterproofing compound for field application to nets. The compound must be susceptible of aqueous application. It must be colorless and not affect the flame-proofing or infra-red reflectance of the nets. It should add as little weight as possible to the fabric, should not seriously affect the "handle" (tackiness, flexibility, etc.) of the net fabrics, and should not adversely affect the tensile strength of the nets.

3. A lightweight material, other than Neoprene, Buna S or other similar rubber substitute now commonly used, which will hold air and CO_2. The material to be of non-critical materials and inexpensive.

4. A light detachable rock drill bit grinder for resharpening detachable bits. Capacity approximately 20 bits per hour and weighing 200 to 250 pounds.

5. Removal of dissolved mineral matter from sea water and brackish water by ionic exchange process.

6. Means of controlling fires in fighting tanks for a sufficient period of time to evacuate personnel. The process should not be injurious to personnel and should be manually controlled and operated.

7. Waterproof compound for treatment of duck used in fabrication of canvas drinking water storage tanks.

8. For field use, a simple, practical and accurate method of determining moisture content of aviator's breathing oxygen.

9. A means for unloading ships by use of quickly erected tramway.

10. Means for increasing life of standard automotive or stationary engines when operated on 91 octane fuel.

11. The use of anti-stripping agents with cut back asphalts to permit effective coating of wet aggregates.

12. Additional methods for the chemical stabilization of soil.

13. A means for removing asphalt from drums and melting to pumpable temperature on a quantity production basis.

14. A quick and effective non-bituminous dust palliative for all climates.

15. Device to maintain or indicate, within 5′, the relation of an aerial camera to the vertical.

16. Reduction of glare from glass surfaces by *durable* coatings suitable for field application.

17. Optical method for determining the difference between an artificial green and a natural green.

18. Destruction and removal of obstacles to landing operations. Obstacles may be visible or concealed and may be off or on shore.

19. Short base, wide angle range finder readily portable and capable of being mounted on a vehicle.

20. Location and destruction of concealed enemy emplacements, pillboxes and similar strong points.

21. Methods of protecting our vehicles from the effects of enemy land mines.

22. Improvements in tank vision devices and control instruments. There is special interest in reducing space requirements and improving performance of gyroscopic compasses.

23. Ingenious and simple decoy devices for purpose of confusing and misleading enemy.

24. Technical data as to strategic enemy targets such as chemical plants, explosive plants, power plants, etc.

25. A voice-transmitting gas mask which would permit the wearer's voice to be heard with clarity.

26. Clothing giving protection against falling pieces of white phosphorous.

27. Colored smokes using readily available pigments for obtaining the desired color.

28. Protective ointments to counteract vesicants.

29. Methods of generating stable artificial fogs, and methods of dispersing artificial and natural fogs.

30. Protection against flame throwers.

31. Odorless toxic agents.

32. Design of life vest which automatically inflates and turns the man on his back when he is thrown overboard by concussion and is unconscious.

33. *Noiseless* hand generator combined with a lightweight flashlight. The generator should be pumped at a rate of 40 revolutions per minute, and the light should be continuously brilliant and start on the first pump.

34. Plumb-bob type of generator to operate a light 6 volt radio. Generator probably operated by a reel winding up and letting out the plumb-bob.

35. Methods of insuring CO_2 inflation of life rafts within 30 seconds time at temperature of — 20 degrees to — 40 degrees F.

MEMBERSHIP.

The following have joined the Society since the publication of the May, 1944, JOURNAL:

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Bailey, R. Cooper, Lieut. Commander, U. S. N. R., Navy Yard, Norfolk, Va.

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Dance, William George, Lieutanant, U. S. M. S., P. O. Box 44, Guilford, N. C.

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